AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 43 No. 565 **APRIL, 1953** PRICE: 3s. 6d.

One of the Actual Drills

16 months continuous production with the same set of EaSiCut drills
The set of 'Easicut' Drills supplied for this 80 spindle Drilling Machine working on Cylinder Heads at Leyland Motors, Ltd., has been in continuous production for over 16 months. Cutting life between re-grinds has also improved considerably—a tribute to ourselves as the manufacturers and Leyland Motors, Ltd., for their choice of a fine drill.
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ENGLISH STEEL CORPORATION LTD
OPENSHAW
MANCHESTER
Makers of high quality Engineers' Cutting Tools for over a century

The performance given by the Easicut drills featured in the above advertisement has been so outstanding that it has been felt worthwhile to draw attention to the following points.

- **No sign of overheating.**
- **Exceptional wear resistant properties at outer corners and chisel edge.**
- **Little or no cratering on cutting faces.**
- **Sharpness at cutting edges maintained.**
- **No adhesion visible on lands.**

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No. 8 VERTICALAUTO

Maximum Swing .. 14¾" dia.
No. of Spindles 6
No of Tooling Stations .. 5

The No. 8 Verticalauto can be arranged for double indexing or dual control A wide range of standard attachments, considerably increasing the scope of the machine, is available.

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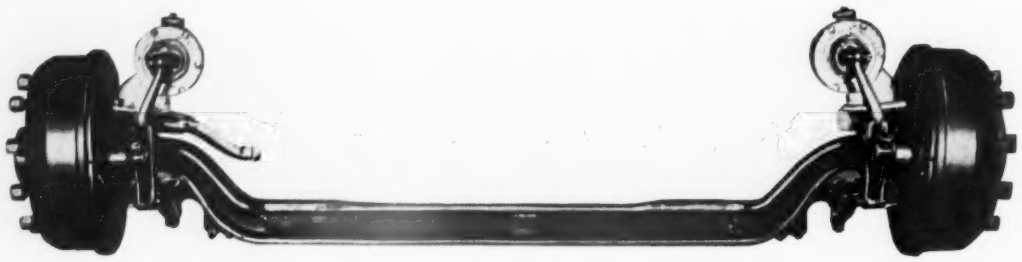

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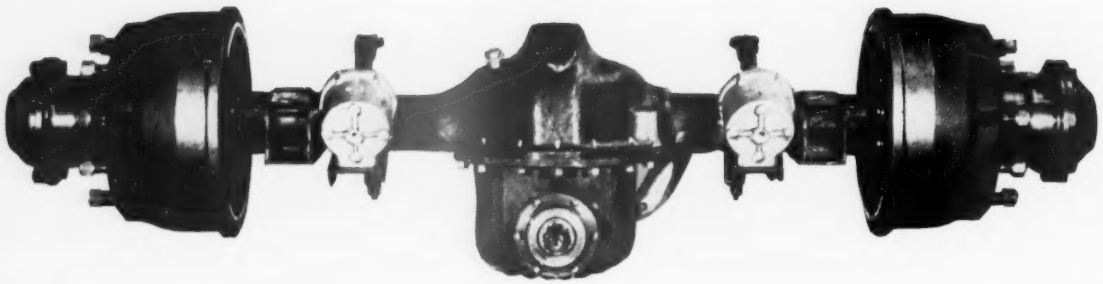

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CYLINDRICAL GRINDING MACHINES

six machines developed from one basic type and unit-built

The R300 series of standard machines, utilising various unit mechanisms in combination, meet a wide variety of production grinding applications in the most economical manner. In addition to these standard types, machines can be supplied with different combinations of mechanisms to meet special conditions.

RG300 PLAIN GRINDER. Basic machine with hand feed to wheel-head and work-table. Suitable for small batch work of simple character.

RE300 PLUNGE GRINDING MACHINE. Employs wide or multiple wheels. Hydraulic movement to grinding head with fast approach, fine feed, and fast return. Dual control axial adjustment for grinding to shoulders. Hydraulic tailstock withdrawal.

RM300 CRANKSHAFT LINE BEARING GRINDING MACHINE. Hydraulic table movement for quick indexing from bearing to bearing. Interlocked controls obviate incorrect operation.

RP300 GENERAL PURPOSE GRINDING MACHINE. Hydraulic grinding head movement and automatic hydraulic table traverse with automatic reversal. Tailstock hydraulically operated.

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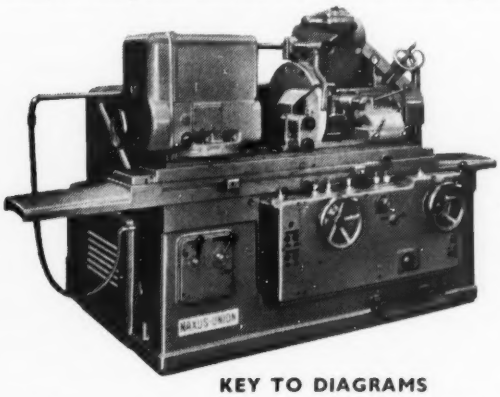
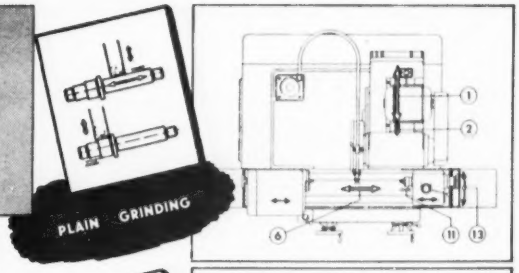
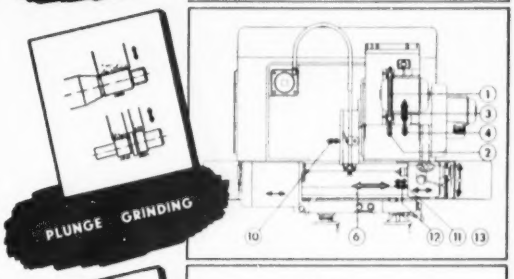
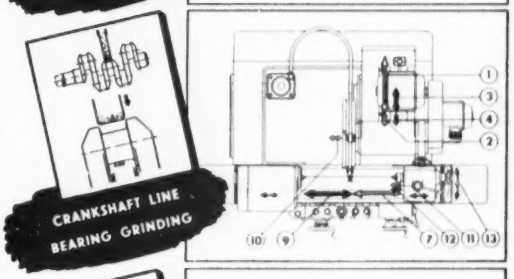
RU300 UNIVERSAL GRINDING MACHINE. In addition to the features of the basic type this model has automatic longitudinal movement of the table, swivelling wheel-head and work-head and internal grinding attachment.

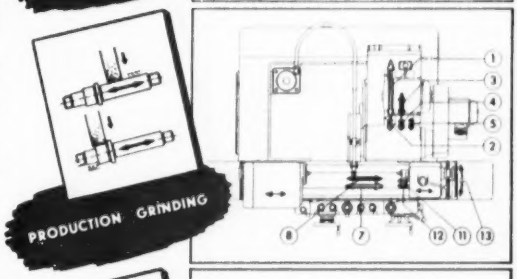


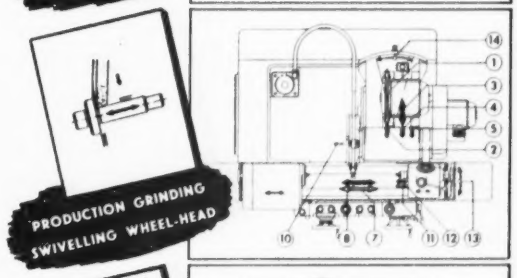
Type RU300 Universal Grinding Machine

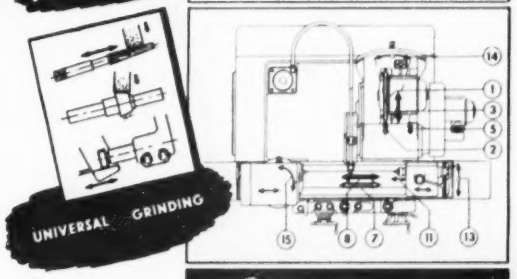












KEY TO DIAGRAMS

BLACK ARROWS Automatic hydraulic movement. **OPEN ARROWS** Hand adjustment.
THIN ARROWS Possibility of adjustment.

1. Hand movement.
2. Fine adjustment by hand.
3. Hydraulic quick adjustment.
4. Hydraulic plunge-cut motion.
5. Automatic fine adjustment.
6. Hand movement, one speed.
7. Hand movement, two speeds.
8. Hydraulic movement with reversal.
9. Hydraulic fast motion.
10. Axial movement.
11. Return motion by hand.
12. Hydraulic return motion.
13. Can be swivelled.
14. Can be swivelled.
15. Can be swivelled.

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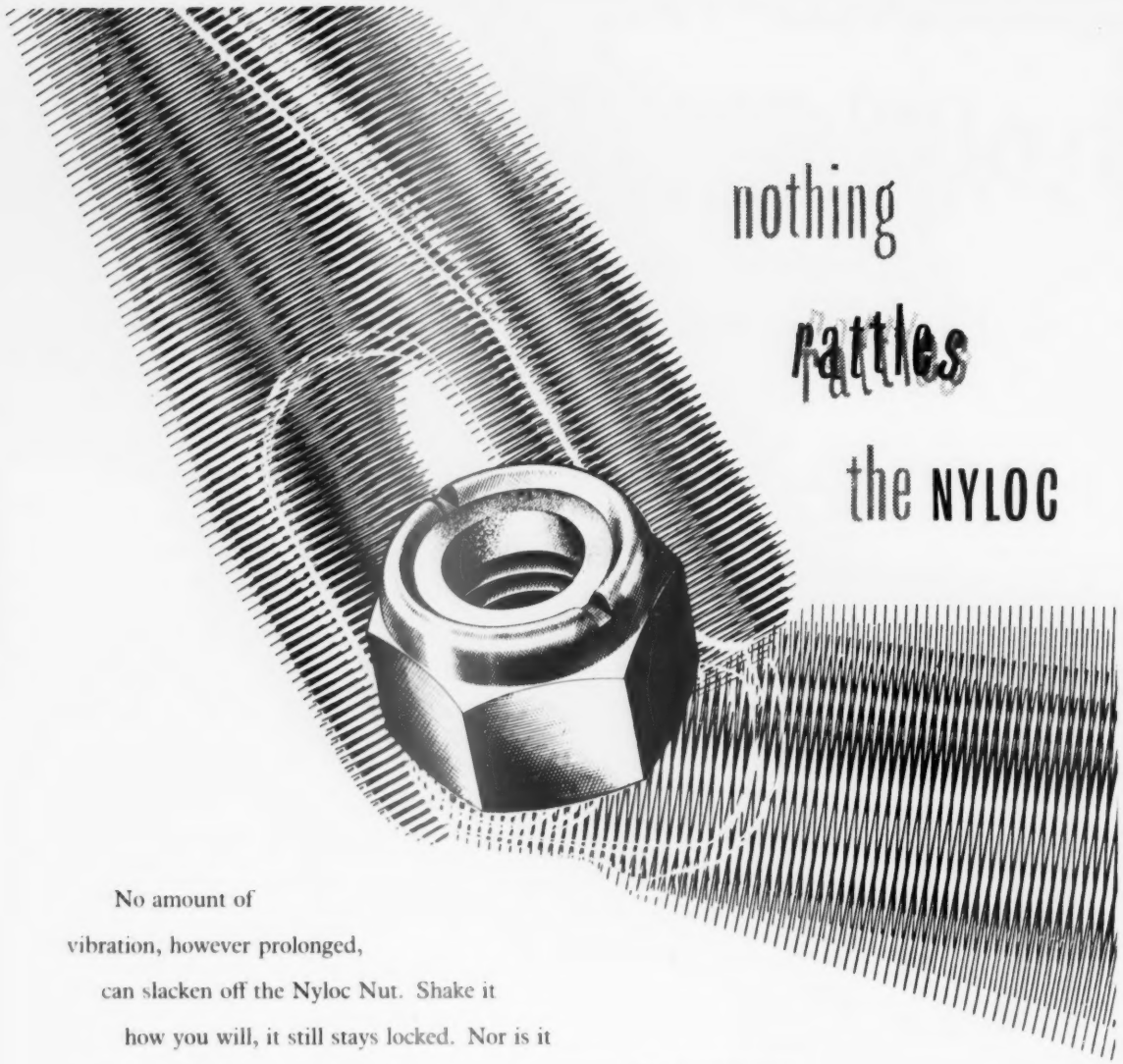

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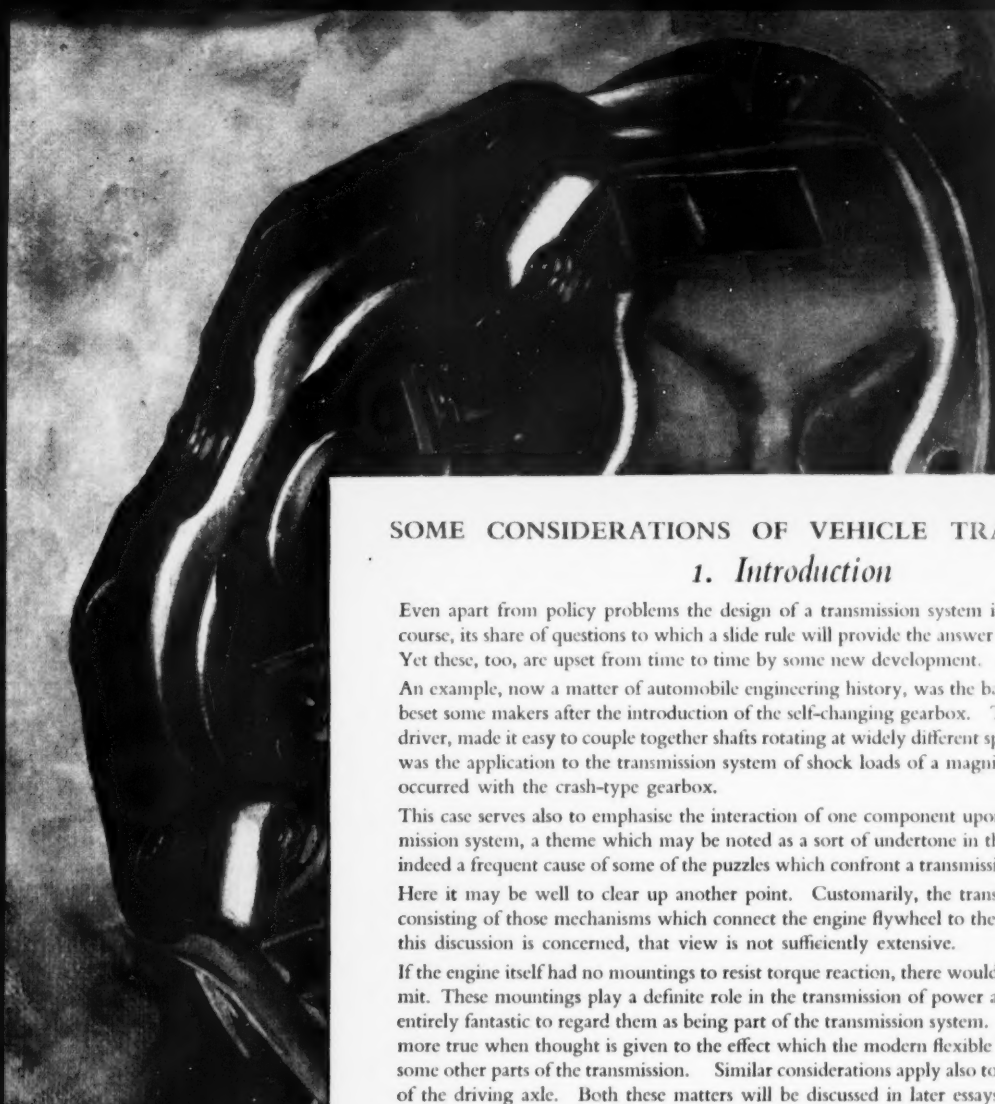

SOME CONSIDERATIONS OF VEHICLE TRANSMISSIONS

1. Introduction

Even apart from policy problems the design of a transmission system is complex. It has, of course, its share of questions to which a slide rule will provide the answer when suitably guided. Yet these, too, are upset from time to time by some new development.

An example, now a matter of automobile engineering history, was the back-axle trouble which beset some makers after the introduction of the self-changing gearbox. This, for the insensitive driver, made it easy to couple together shafts rotating at widely different speeds. A consequence was the application to the transmission system of shock loads of a magnitude which had rarely occurred with the crash-type gearbox.

This case serves also to emphasise the interaction of one component upon another in the transmission system, a theme which may be noted as a sort of undertone in these short essays. It is indeed a frequent cause of some of the puzzles which confront a transmission engineer.

Here it may be well to clear up another point. Customarily, the transmission is regarded as consisting of those mechanisms which connect the engine flywheel to the final drive. So far as this discussion is concerned, that view is not sufficiently extensive.

If the engine itself had no mountings to resist torque reaction, there would be no power to transmit. These mountings play a definite role in the transmission of power and it is, therefore, not entirely fantastic to regard them as being part of the transmission system. That becomes all the more true when thought is given to the effect which the modern flexible mounting exerts upon some other parts of the transmission. Similar considerations apply also to the suspension springs of the driving axle. Both these matters will be discussed in later essays.

For the moment, let them serve to introduce another aspect of our subject. Flexible suspension of the engine was evolved to meet a demand for greater refinement in the general behaviour of the car. It cushions the torque-reaction kick from the modern high-compression engine, and absorbs much of the vibration innate in every reciprocating engine. In so doing, it also smothers at birth various noises which otherwise would develop from the sympathetic resonance of body panels and other parts.

In this discussion, a fact to be remembered is that the demand for greater refinement concerns every part of the vehicle, including the transmission system. Vibration in this department can be just as damaging as when it arises elsewhere. Moreover, although a little may be possible in that way, it is impracticable for suspension reasons to isolate the entire transmission system by means of flexible mountings.

There is thus a pressing need to ensure (so far as may be possible) that the transmission system is not a source of vibration or other disturbance of this general type. Vibration is, of course, commonly associated with periodic or cyclic accelerations and decelerations. Anything of that sort, therefore, must be avoided or reduced to a minimum.

Where cyclic speed variations are inevitable—and it must be admitted that they are virtually inherent in most active mechanisms—they must be absorbed, if possible, so that they shall not exercise their evil influence on neighbouring components.

As will appear in the remainder of this series, there are in the transmission system all too many opportunities for vibration and other mechanical disturbances. These effects, if ignored by the designer, could cause much more than lack of refinement. Mechanical failures are well within their range of achievement. It is therefore essential to recognise them and to cope with them.

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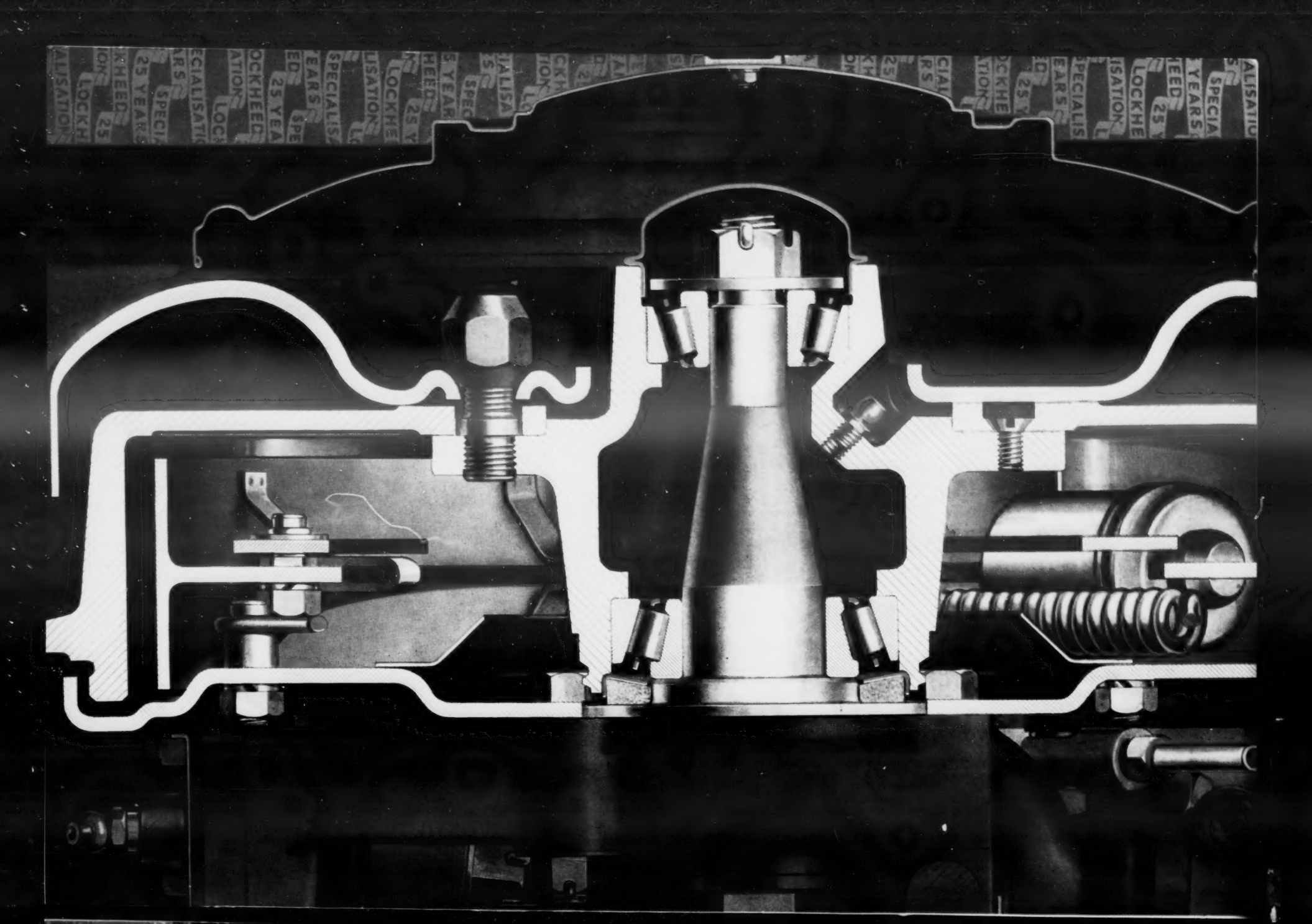
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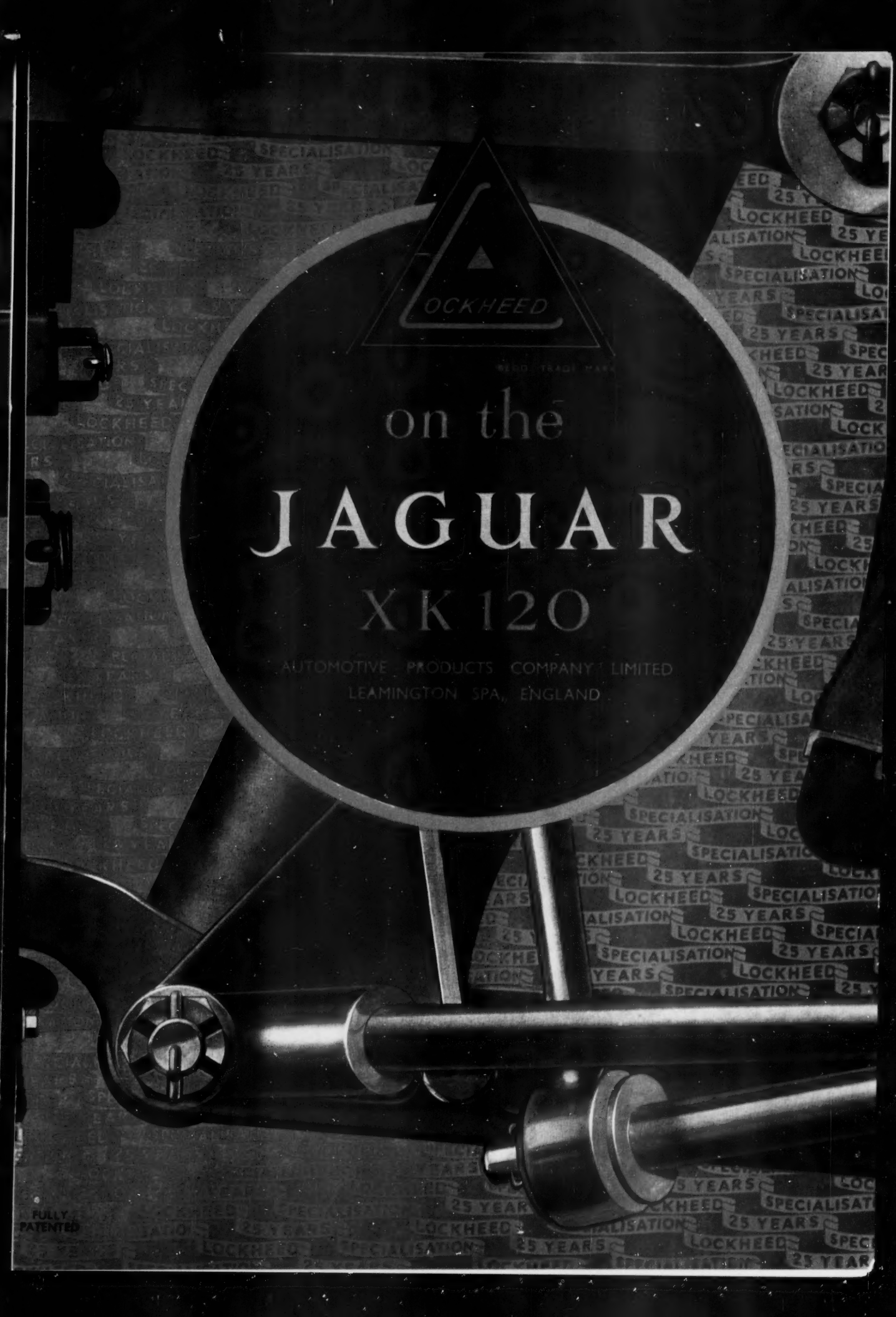
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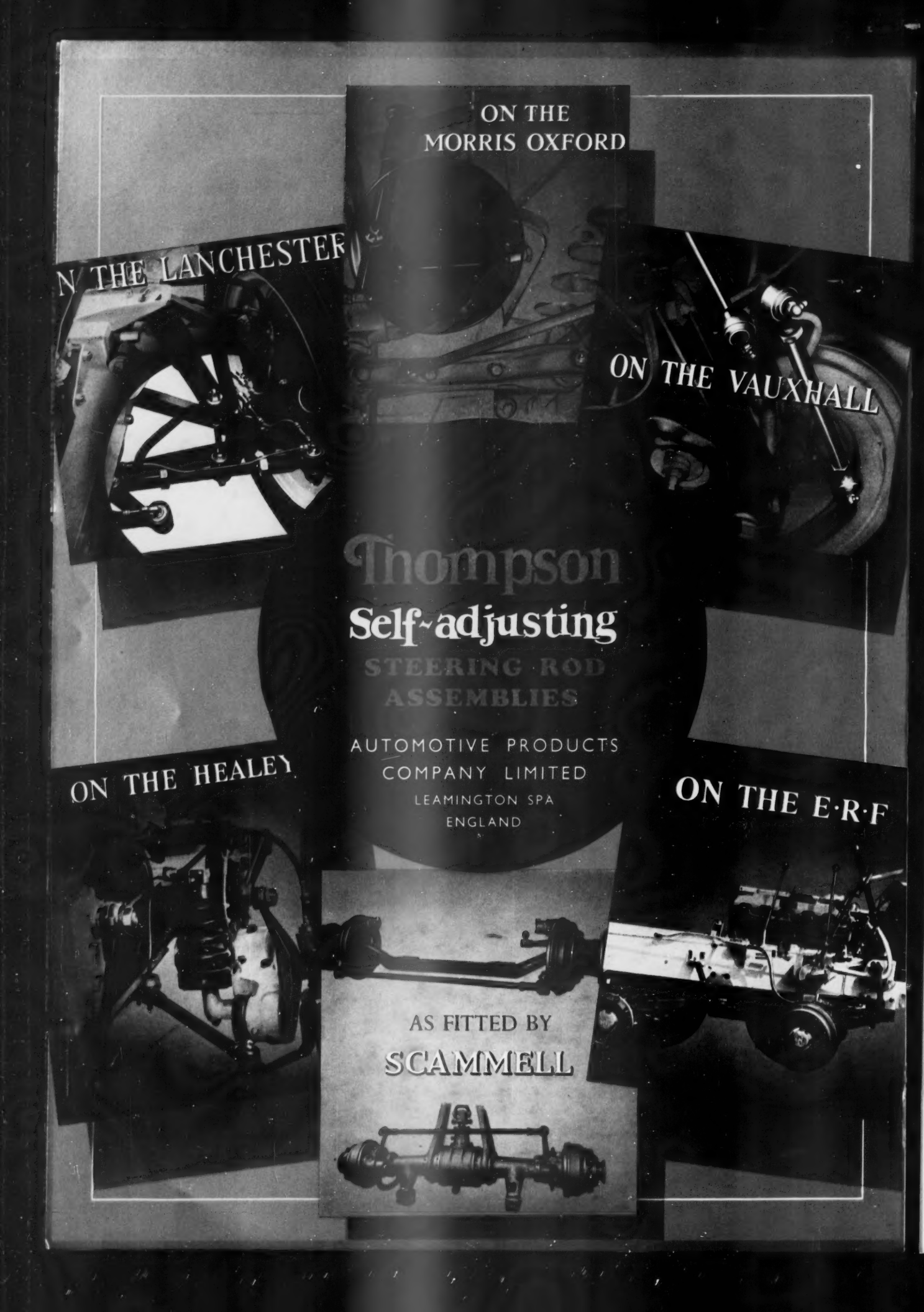
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"Only one way, so far as I'm concerned."

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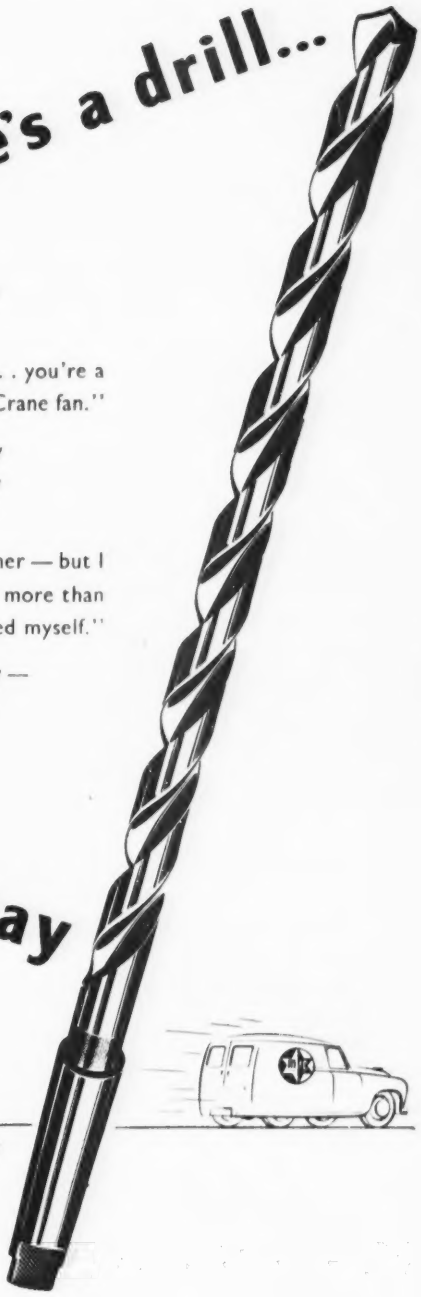
"Who wouldn't be? . . . They work miracles — particularly with the specials."

"I'm only a learner — but I must say, I'm more than convinced myself."

"You can't go wrong, old boy — twist drills or cutting tools, it's all the same to them."

...there's a way



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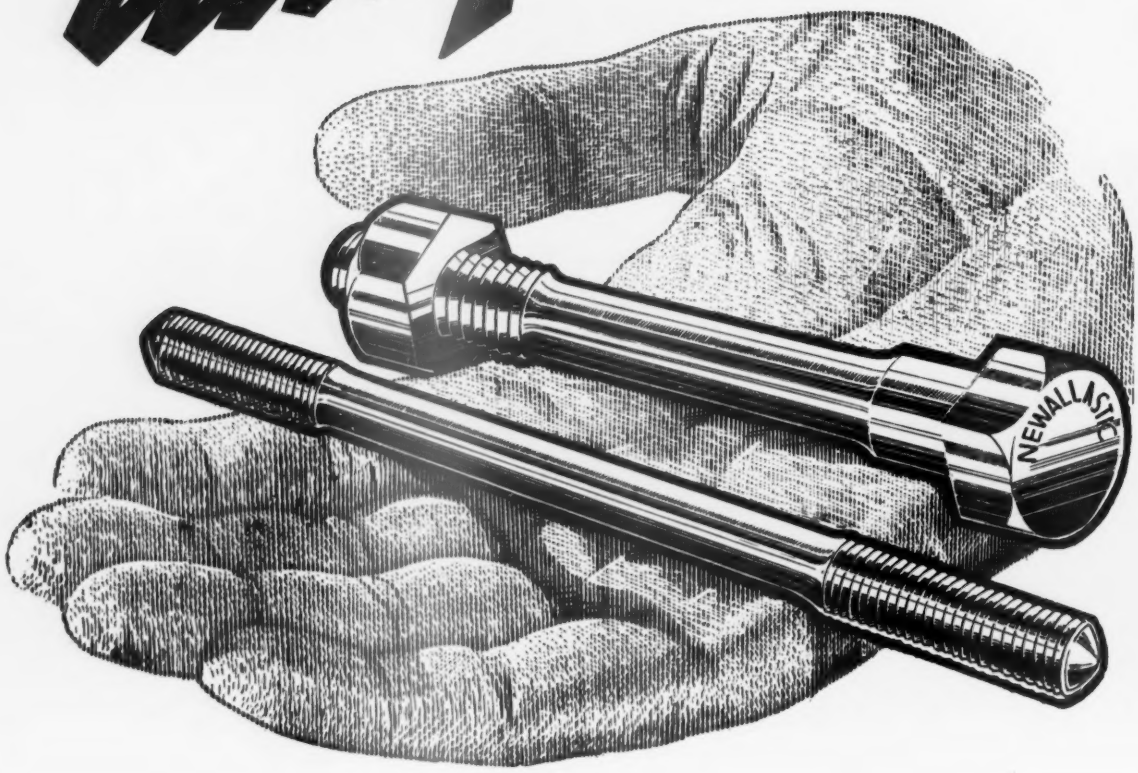

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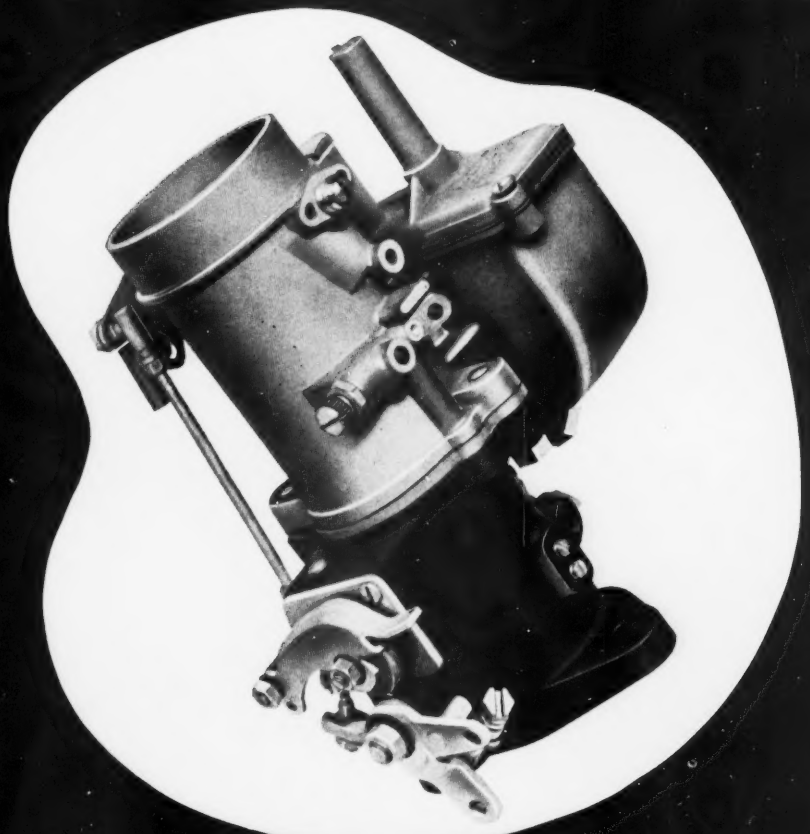
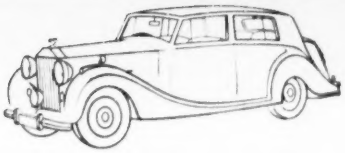
POSSILPARK GLASGOW · N

THINK ZINC





Almost as quickly as you can say the words "Think Zinc", molten zinc alloy has become the Stromberg carburettor you see here. This speed in production is just one reason why the automobile industry—like so many other industries—has adopted the zinc alloy die casting process. Other outstanding advantages are:—
SCOPE FOR DESIGNER · EXTREME ACCURACY · GOOD CORROSION RESISTANCE · STABILITY STRENGTH · LONG LIFE OF DIES · LITTLE NEED FOR MACHINING · LOW UNIT COST
The Association welcomes inquiries about the use of zinc alloy die castings. Publications and a list of Members are available on request. We suggest you write for our booklet "Zinc Alloy Die Castings and Productivity."
Zinc is now plentiful. There are no restrictions on its use.

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Think of a diesel's life

Discussing the performance of Messrs. Day's Leyland Octopus are (left to right): Mr. G. Forrest, driver; Mr. J. L. Pittard, Day's Transport Manager; and Mr. A. G. Chase, of Shell-Mex and B.P. Ltd.

then double it!

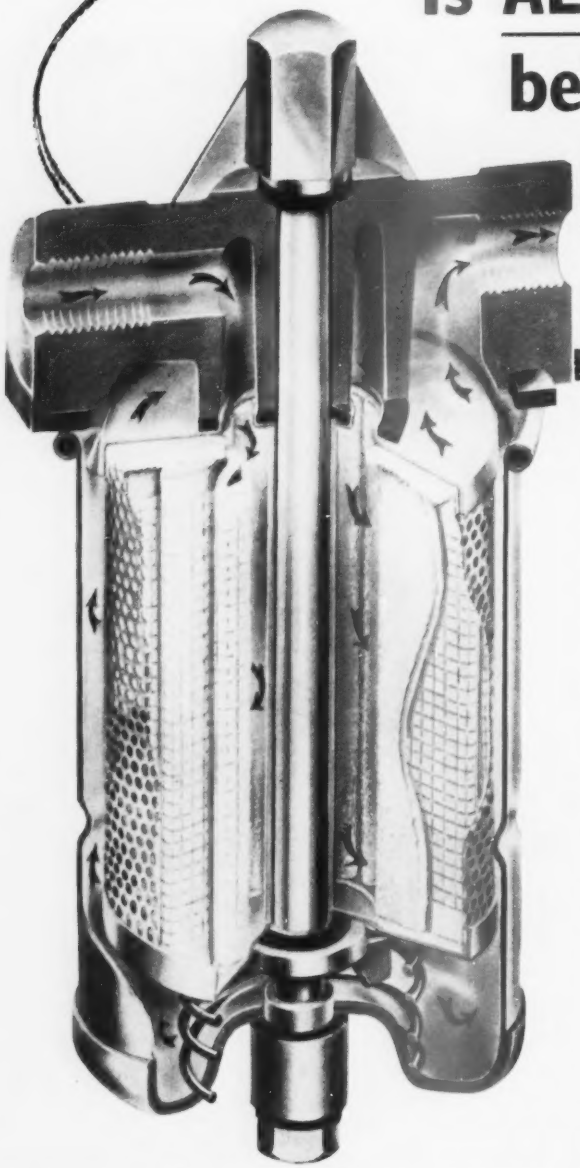
This Leyland Octopus, owned by Messrs. G. A. Day Ltd., builders' merchants, of Portsmouth, has done over 220,000 miles of day and night work in just over 2½ years. Apart from normal servicing, all that has been done is to draw the pistons and fit new rings. Sump oil is changed every 3,000 miles.

Shell Rotella Oil has been used exclusively in this hard-worked Leyland Octopus which averages 280 miles per 24-hour day. The new Shell Rotella Oil—the result of years of specialised research—embodies all the superior qualities of the former oil but with even greater anti-ring stick and anti-wear properties.

SHELL ROTELLA OIL

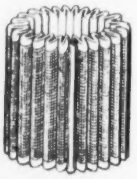
SHELL LEADERSHIP IN LUBRICATION

Is ALL THE OIL being filtered . . . ALL THE TIME?



That is the vital question which must be asked when considering filtration claims. Engineers today accept the fact that for real protection, partial flow filtration is not enough. But remember! even a so-called full flow filter can allow irreparable damage if the by-pass comes into operation for long periods when the oil is cold.

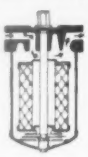
CONSIDER THE FAMOUS VOKES ELEMENT



Here is the VOKES filter element—unchanged in principle because thirty years of constant research have found nothing better. By its principle of "filtration-in-depth" it avoids the possibility of particles being driven straight through the filtering medium and also ensures a slower build up of surface sludge.

This then is the vital point. If a so-called full flow filter cannot cope with the oil when it is cold and sends it direct into the engine, then it is not filtering all the oil all the time. Consider, too, the question of "cleanability". This is highly important where plant and machinery is operating in remote areas, when the non-availability of expendable elements might put the equipment out of service. Also, although the VOKES element may cost a little more in the first place, the fact that it can be cleaned several times gives long-term economy.

The VOKES direct flow device is a safeguard to the engine and not just to the filter itself. It comes into operation only under *abnormal* conditions and *not* for long periods merely because the oil is cold. When it is called upon, however, it is truly foolproof. Illustration (left) shows flow-through filter element. Illustration (right) shows by-pass flow.



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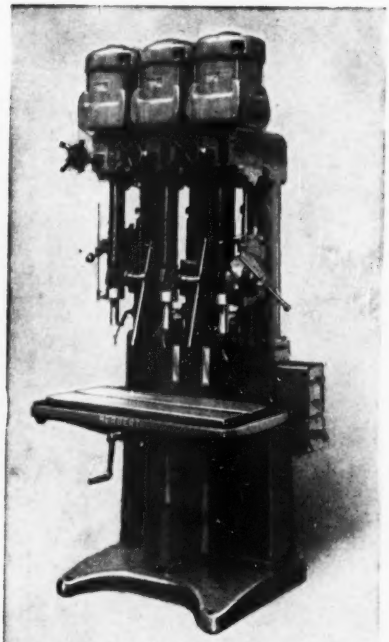
FOR HIGH-PRODUCTION DRILLING OF HOLES FROM THE SMALLEST, AT 18000 r.p.m., UP TO 1¼in.

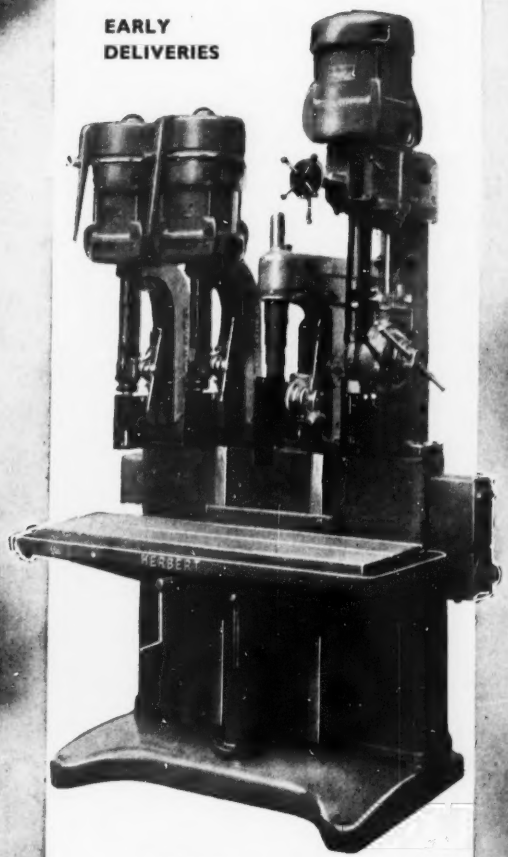
WIDE RANGE OF STANDARD MACHINES AVAILABLE. UNIT CONSTRUCTION ENABLES SPECIAL MACHINES TO BE BUILT UP TO SUIT ANY SPECIFIC WORK.



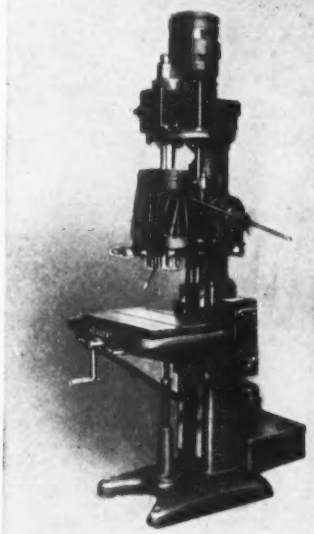
TYPE V THREE SPINDLE FITTED WITH ONE GEARED AND TWO PLAIN SPINDLES AUTOMATIC REVERSE FOR TAPPING IS FITTED TO THE LEFT-HAND SPINDLE.

TWO TYPE C TOP COLUMNS MOUNTED ON A FOUR-SPINDLE BASE. LEFT-HAND COLUMN IS ADJUSTABLE AND A TELESCOPIC CONNECTION WHICH ENABLES SPINDLES TO BE FED SIMULTANEOUSLY IS FITTED BETWEEN THE SPINDLES.

EARLY DELIVERIES

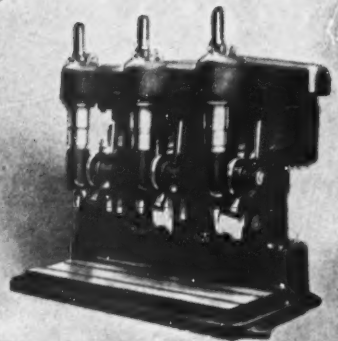


COMBINATION OF TOP COLUMNS MOUNTED ON A SINGLE BASE. SPEED RANGE 74 TO 5600 r.p.m.



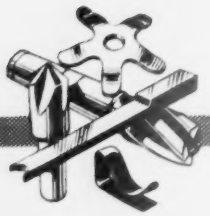
TYPE M MULTI-SPINDLE

TYPE B BENCH



TYPE H THREE-SPINDLE BENCH

ALFRED HERBERT LTD
COVENTRY



PAN CONVEYOR FURNACE

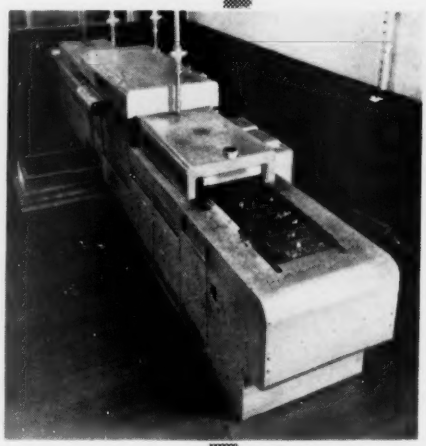
The pan conveyor furnace illustrated is used for hardening small components which are placed on pans fitted together by chain links, the components passing through the water cooled zone into the heating chamber and are then automatically plunged into the quench tank.

The heating chamber is 5 foot long and has a cross section of 20" x 6" high, built of refractory material with specially shaped hard bricks to support the large diameter nickel-chromium tubular elements placed in the roof and hearth. The quench chute is also provided with elements.

The top part of the furnace chamber is removable allowing access to the heating elements, conveyor and floor tray. The quench opening is provided with a downward extension for sealing into the quench tank and is fitted with a gas union for feeding gas to the interior of the chamber.

The heating chamber is split into two zones and has a maximum operating temperature of 900—950° C.

The coolers each end of the heating chamber are features which make Wild-Barfield Pan Conveyor Furnaces very efficient. The cooler at the exit end is for cooling the work trays in an atmosphere to avoid scaling which would be detrimental to the finish of the work, as during heating they would give off oxygen from their scale which would attack the work. A further function of this cooler and the one at the entry end is to give a satisfactory balance of atmosphere in the furnace, by contracting the gas before it leaks out at the ends.



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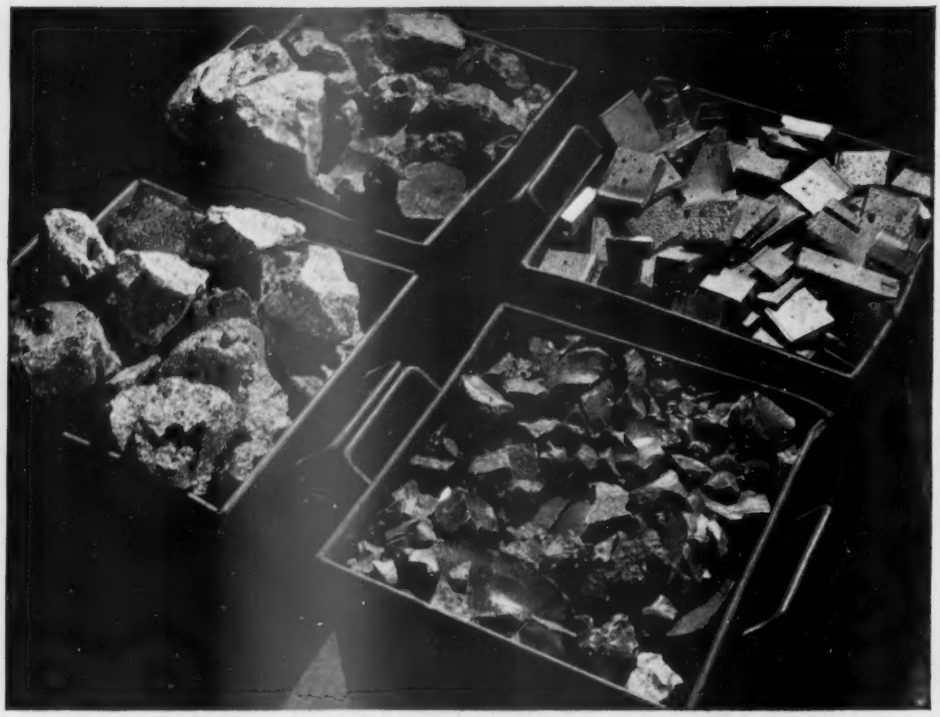

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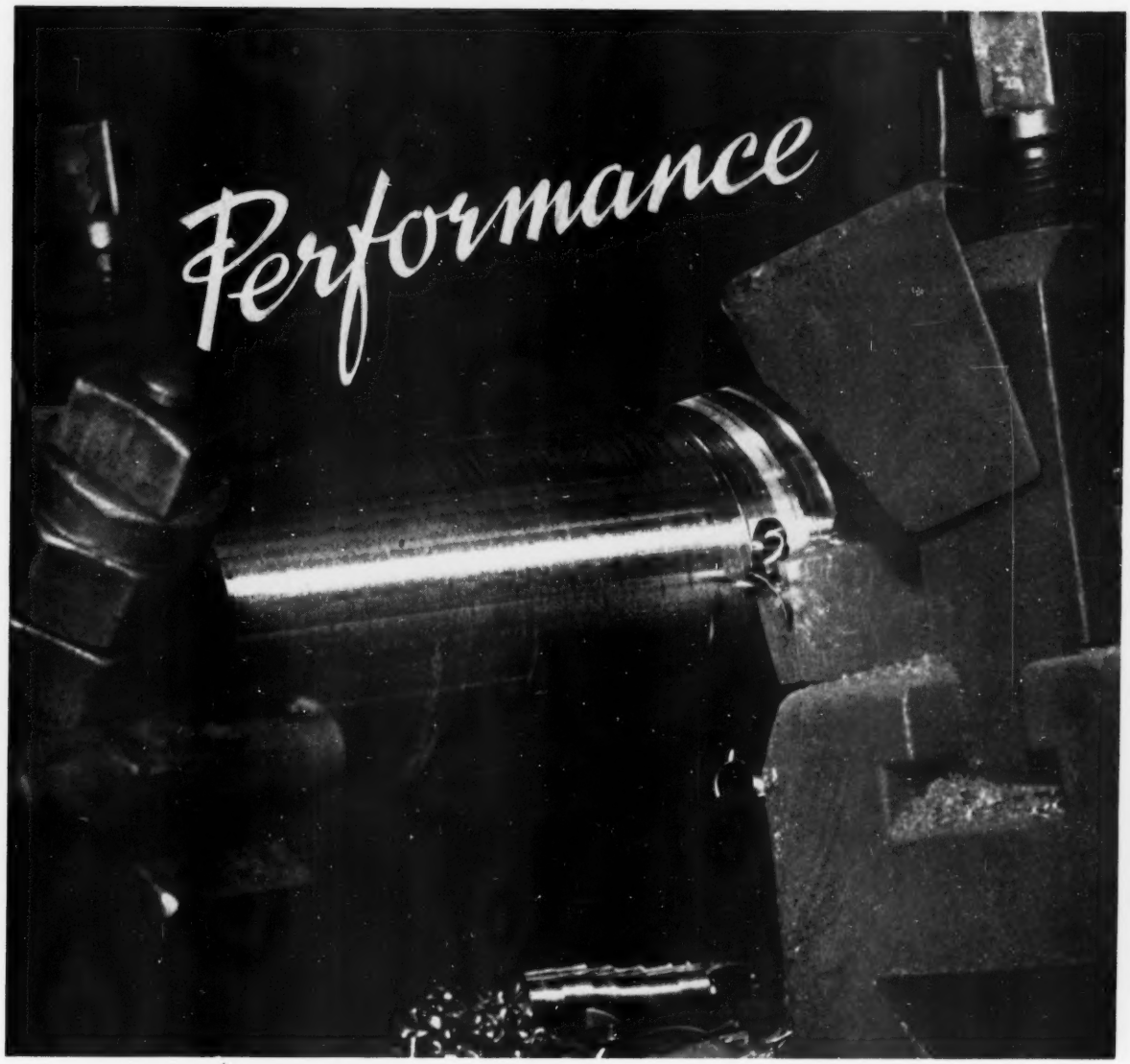

The above photograph shows a Cutanit Grade R tool in a routine operation, machining a hard steel motor shaft at 350 ft/min with a 7/8″ depth of cut.

Photograph by courtesy of Messrs. Metropolitan-Vickers Electrical Co. Ltd.

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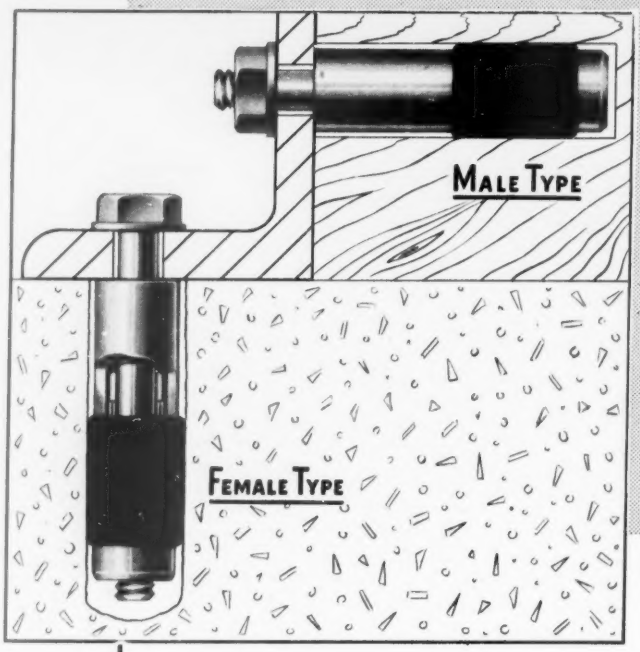

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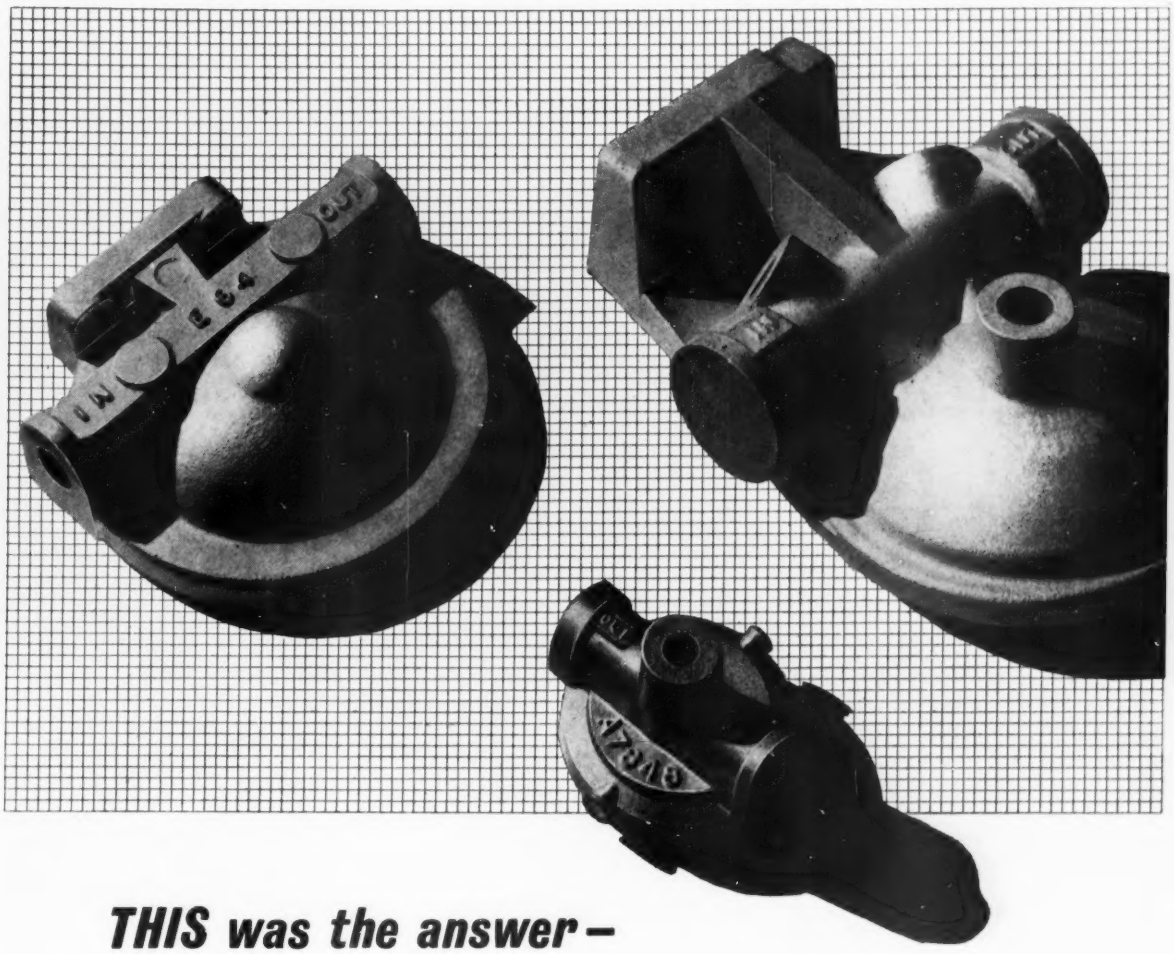

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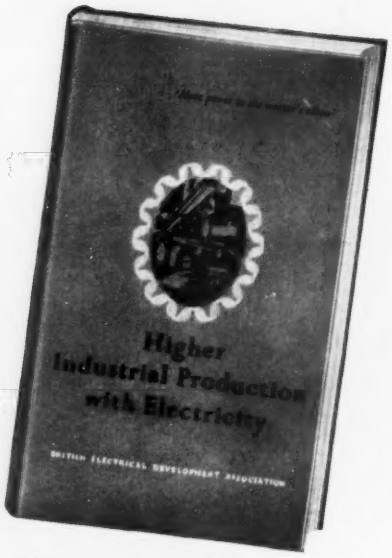


INCREASED PRODUCTIVITY is the chief task facing Industry today, and there are many ways of achieving it. The solution may lie in higher speeds, or more exact control, or more refined methods of heating, or the latest system of inspecting for quality. But in every case, Electricity has an important part to play.

The British Electrical Development Association has published "Higher Industrial Production with Electricity" to help in this vital task. The book is written for management, and for all who are concerned with industrial planning and processes.

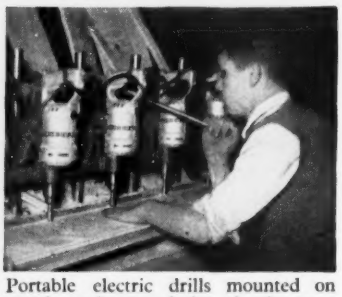
There are chapters on Lighting, Motors and Control Gear, Furnaces and Heat Treatment, Welding, Handling, and Inspection. Make a note to get a copy.

Now ready

"Higher Industrial Production with Electricity", some illustrations from which are shown, is available at 8/6 (9/- *post free*) *from The British Electrical Development Association, 2 Savoy Hill, London, W.C.2 and from your Electricity Board.*

An electric stud-welding gun in operation on a dust collector component.

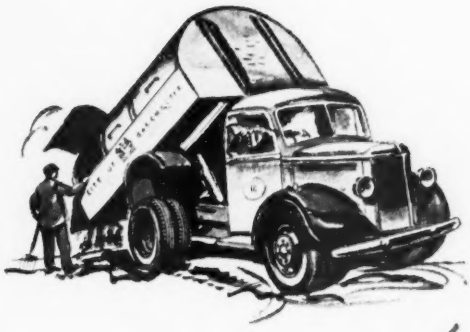
Left: an electronic control cubicle with one unit withdrawn. Right: an electric skip hoist charger for a new blast furnace in South Wales, where the entire hoisting and charging process is automatic — initiated by the pressing of a single button.



Portable electric drills mounted on stands and ganged for simultaneous operation in a Midlands Joinery shop.

Electricity for PRODUCTIVITY

Hydraulic efficiency?



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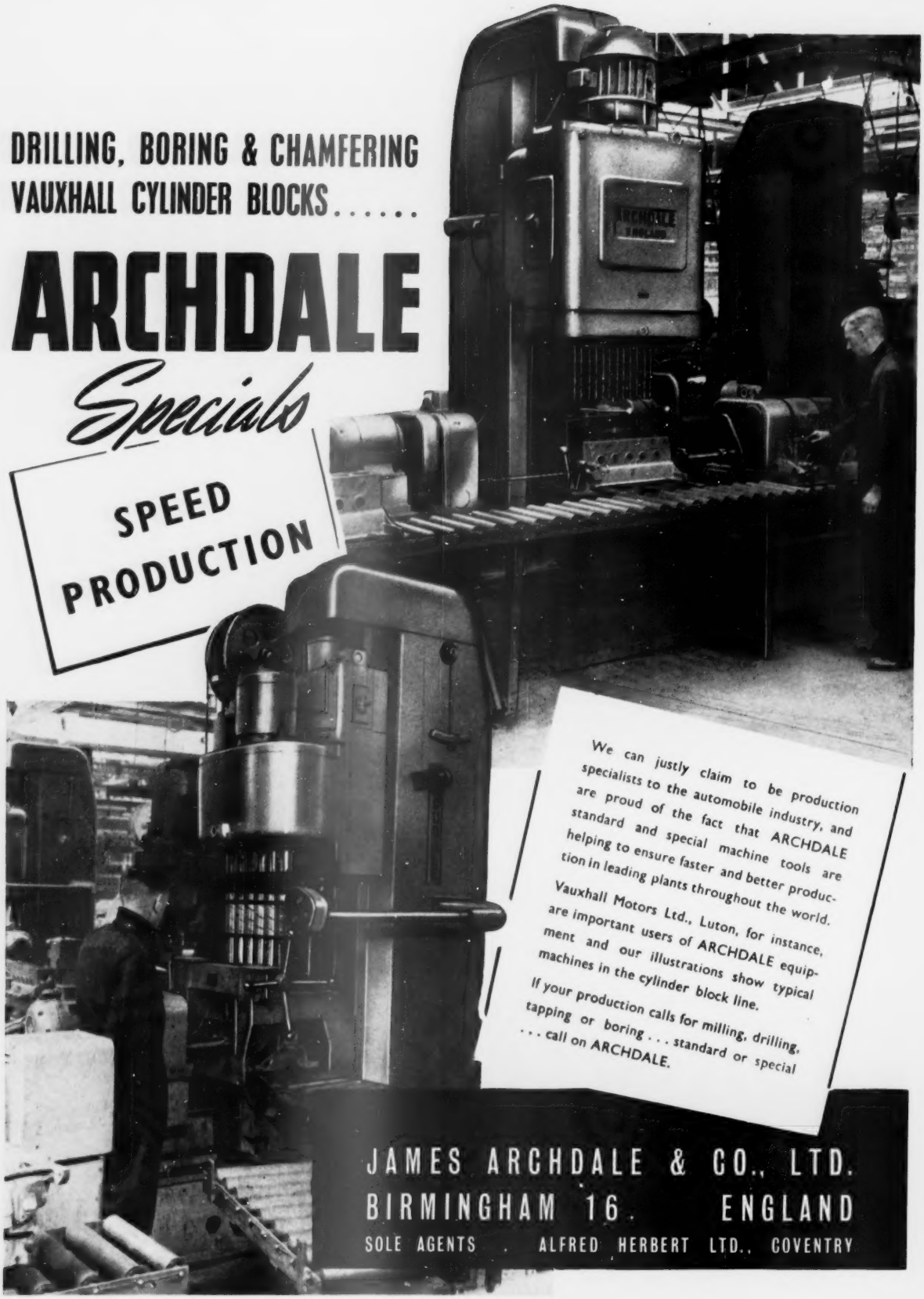

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Vauxhall Motors Ltd., Luton, for instance, are important users of ARCHDALE equipment and our illustrations show typical machines in the cylinder block line.

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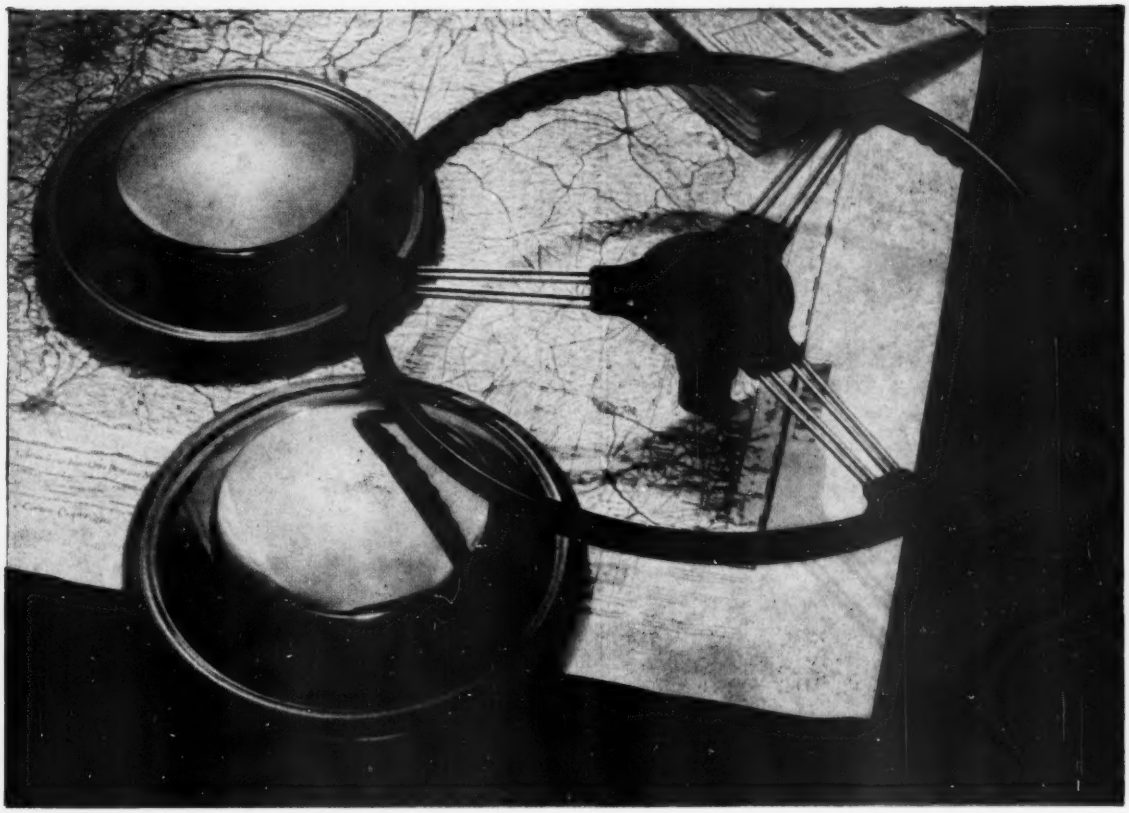

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Our colour photograph shows Hub caps by courtesy of the Ford Motor Company. Steering Wheel by courtesy of Wilmot Breeden Limited.

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Page One
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Why YOU should fit the M.M.D. Clevis . . .

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Positive linkage allows a more positive gear change and the M.M.D. Spring Loaded Clevis Joint is the perfect solution to the problem of "sloppy" remote control gear change assemblies. Wear, slack and rattle on *all* mechanically operated linkages, can be eliminated throughout the life of the car, with the aid of this rationalized clevis design. Available to suit all requirements of the Motor Industry.

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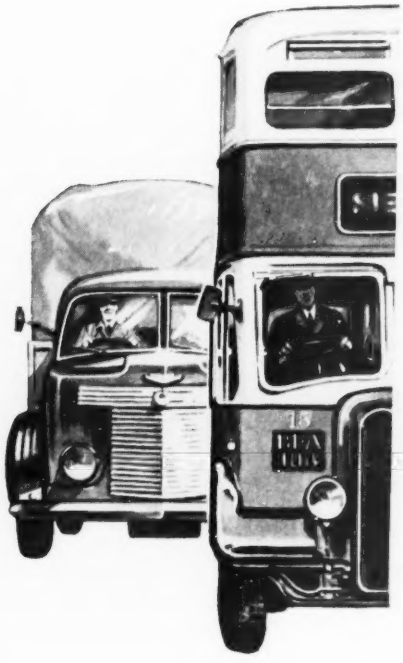
Soon pays for itself!

GIVES YOU 10% EXTRA SERVICE FROM EVERY VEHICLE!



The ' Syndromic ' system lubricates up to 40 bearings —at periods automatically controlled by mileage— *while your vehicles are on the road!* In this way, hold-ups for lubrication are eliminated and you get as much as 10% more service out of your fleet at no extra cost. In fact, ' Syndromic ' automatic lubricators are *more* economical. They save 75% on lubrication, reduce wear on bearings, etc., and therefore cut maintenance and replacement costs to a minimum.

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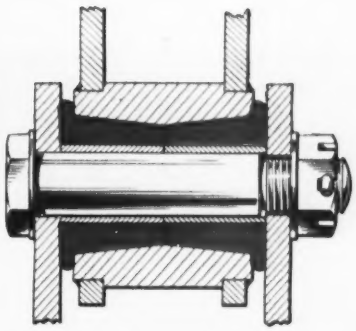
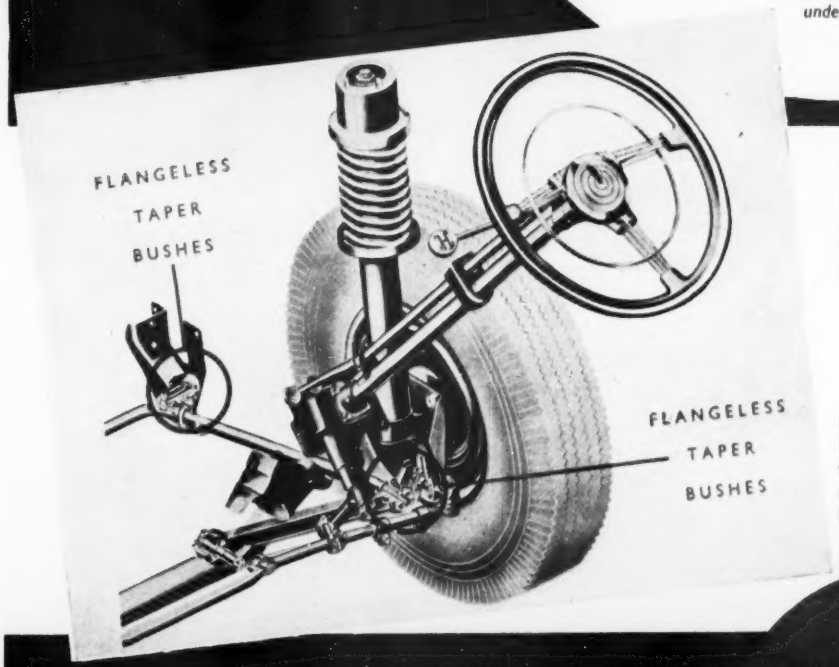
FLANGELESS TAPER BUSHES for I.F.S. systems



Scientific Development Cuts Costs

Constructed on the Silentbloc principle with a self-forming flange for extra-snug fitting and less wear these bushes ensure maximum efficiency at lowest cost.

Right: Flangeless Taper Bush sectioned to show construction. In the housing flanges develop under compression.



★After exhaustive testing Ford Motor Company Ltd. now fit Silentbloc Flangeless Taper Bushes in the I.F.S. of the "Consul" and "Zephyr Six".

Left: Front Suspension of the "Consul" and "Zephyr Six".

Flangeless Taper Bushes and Frustacon mountings provide the first complete and scientifically designed rubber insulation between wheel and body.

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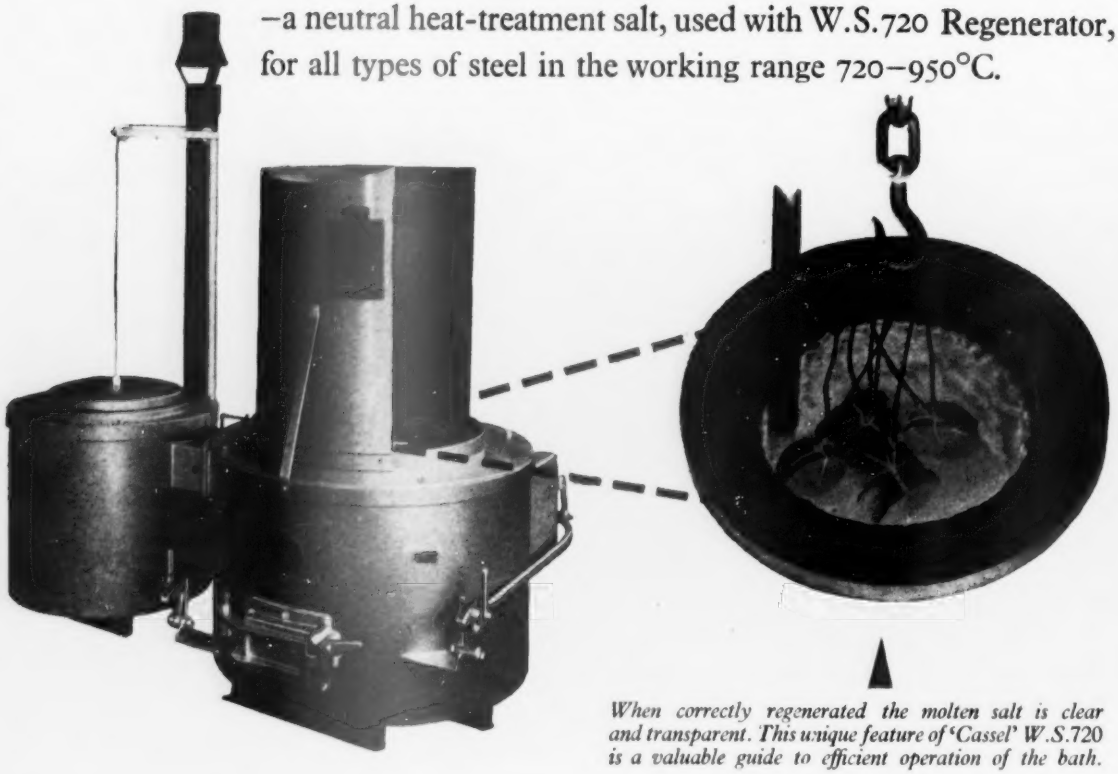
AC-DELCO DIVISION OF GENERAL MOTORS LTD., DUNSTABLE, ENGLAND and Southampton, Hants.

'Cassel' W.S. 720

–a neutral heat-treatment salt, used with W.S.720 Regenerator, for all types of steel in the working range 720–950°C.



When correctly regenerated the molten salt is clear and transparent. This unique feature of 'Cassel' W.S.720 is a valuable guide to efficient operation of the bath.

- **Neither carburises nor decarburises even under the severest conditions.**
- **Readily water-soluble.**
- **Ideal for carbon and alloy tool steels and carburised parts that have been selectively machined.**
- **A very fluid bath of low density resulting in low salt consumption.**

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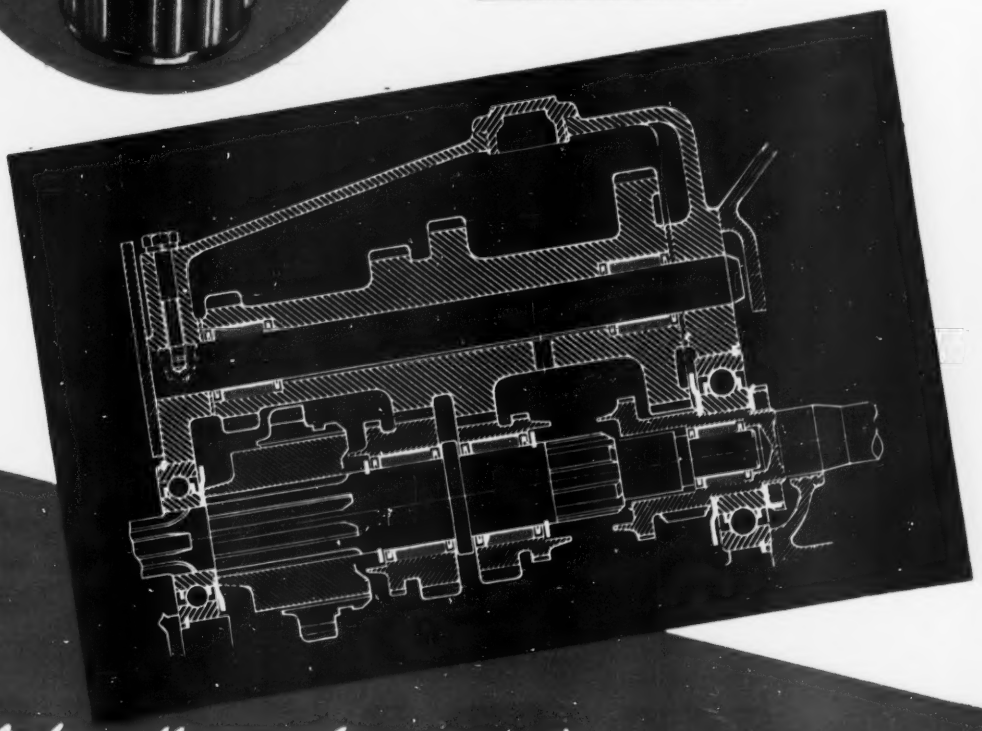

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MOULDED BRAKE LININGS

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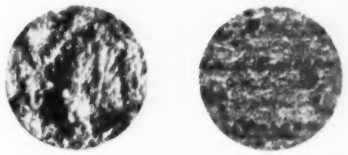
it takes a *moulded lining* to come as clean as this—

—one good reason why every CAPASCO Lining is moulded

Every trace of grease can be removed by normal degreasing methods. CAPASCO Moulded Linings are impervious to oil and grease mainly due to their uniform texture which is illustrated by the micro-photographs. Woven Linings are absorbent and should be discarded but CAPASCO can be cleaned and put back into service with the assurance of full efficiency. In addition, the other special features of CAPASCO are:— Dimensional Stability—High Mechanical and Impact Strength—Rapid 'Wet' Recovery—Extreme Resistance to Frictional Fade—Uniform Wear to 'Wafer' Thickness—Non-Abrasive to Brake Drums—Suitable for Medium and Heavy Duty Application involving High Temperatures.



These micro-photographs of a Woven Lining and of a CAPASCO Moulded Lining show the very striking difference in texture. It is this homogeneous structure of a Moulded Lining which is basically responsible for the excellent overall characteristics.

CAPASCO

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Ransome & Marles

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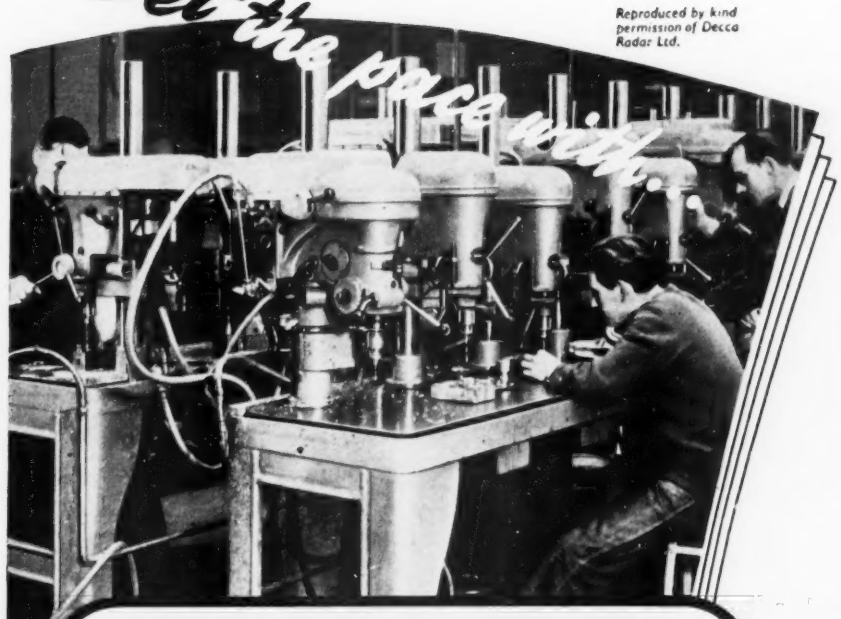
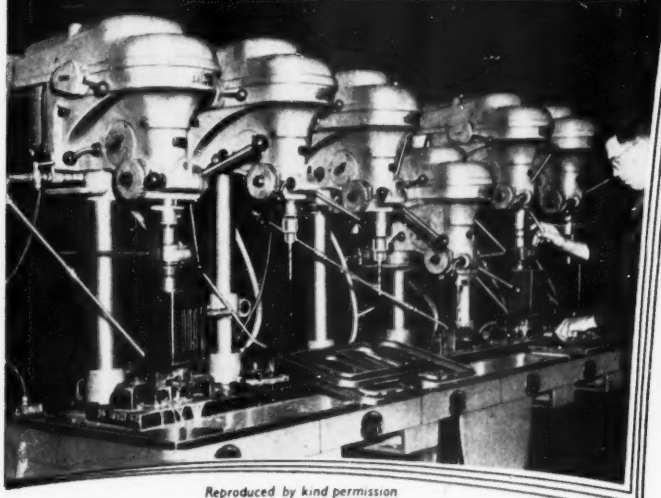


'PACERA'



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Pacera Drilling Machines (from ¼" to 1½" capacity) are manufactured at our own up-to-date specialised plant at Slough

W. J. MEDDINGS LIMITED

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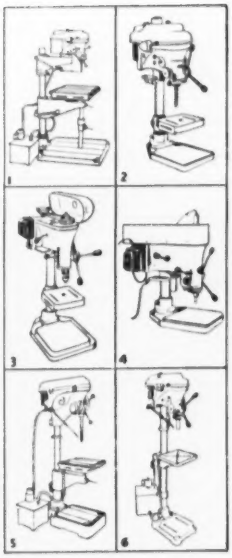
Telephone: MAYFAIR 6417



1. ARTICULATED MODEL
2. MODEL M.B.4 (¼" capacity)
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4. GLASS DRILLING MACHINE
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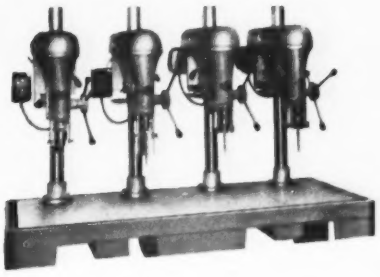
from 2 to 200 COMBINATIONS — with 'PACERA' MULTIPLE HEAD UNITS!

DESIGN YOUR OWN MULTIPLE DRILL COMBINATION WITH PACERA SECTIONAL TABLES

Heads available in a wide range of speeds from 80 to 9,000 r.p.m.—¼" to 1" capacity

There is a Pacera tool for your job. . .



. . . but they're a subject to which Glacier have devoted thirty years, a large research organisation, a first-rate plant and a quite exceptional labour force. World-renowned engines use Glacier Bearings. Thin-wall and thick-wall too . . .

GLACIER
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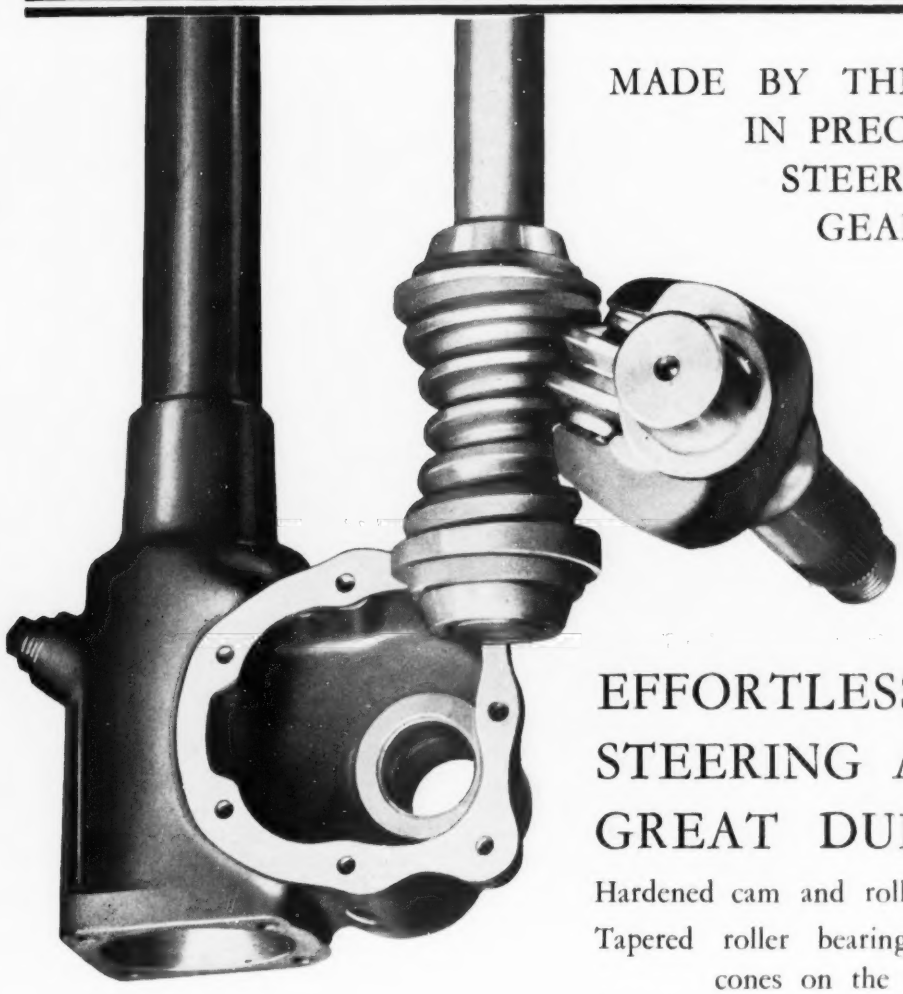
stands for **all** plain bearings

THE GLACIER METAL COMPANY LIMITED, ALPERTON, WEMBLEY, MIDDLESEX
MAKERS OF GENUINE THIN-WALL BEARINGS AND THICK-WALL BEARINGS



MADE BY THE PIONEERS
IN PRECISION
STEERING
GEARS

EFFORTLESS STEERING AND GREAT DURABILITY

Hardened cam and roller.

Tapered roller bearings (with detachable cones on the larger sizes).

Double bearing support to rocker shaft.

End-location adjustable.

Larger angular movement.

Very compact box.

For fore-and-aft or transverse layout.

Trunnion or spigot mounting alternative on heavy types.

THE 'MARLES' DOUBLE-ROLLER GEAR, made in a range of sizes covering every type of chassis.

MARLES

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ENGINEERING CO. LTD., DALLOW ROAD, LUTON

Sole proprietors of the Marles Steering Company Ltd.

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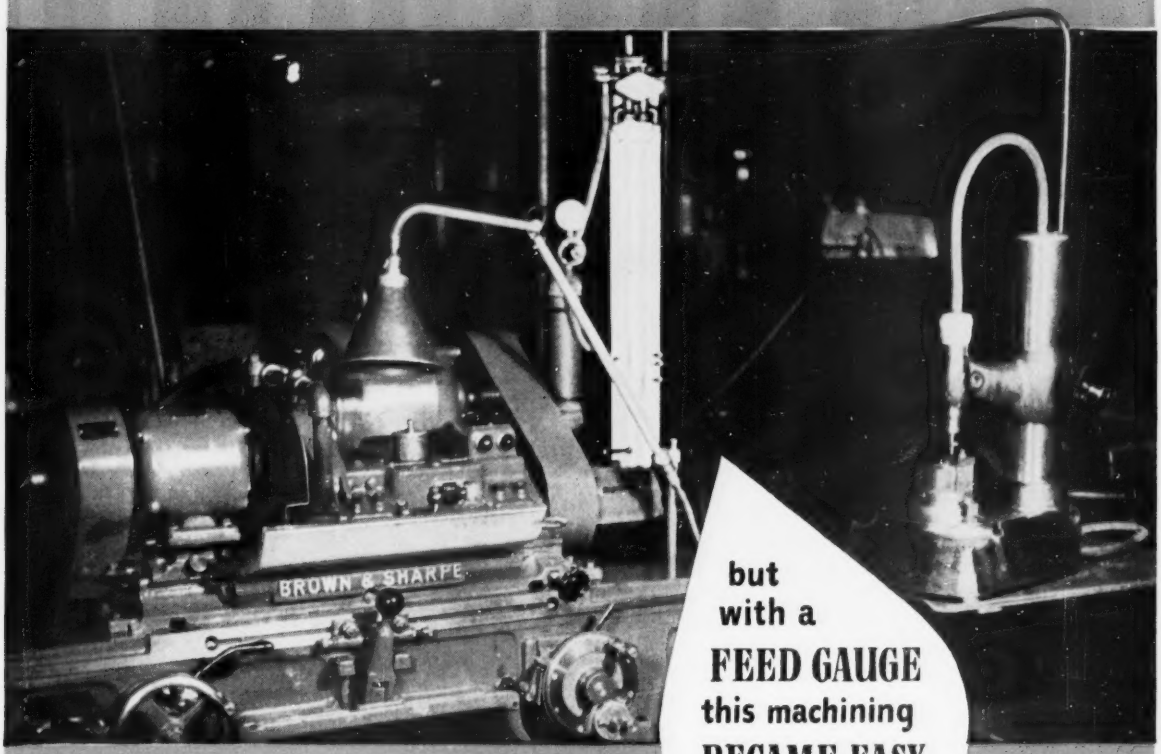
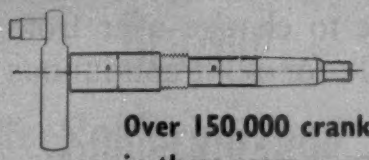
RUBERY OWEN & CO., LTD., DARLASTON, SOUTH STAFFS. TELEPHONE: DARLASTON 130.

It doesn't seem possible –



but
with a
FEED GAUGE
this machining
BECAME EASY



Over 150,000 crankshafts in three years

Tolerance on two diameters only ± ·00006" (avoiding form errors)

Rejects for dimensional causes negligible : under 0·1%

Entirely unskilled operator

No inspection before assembly

For fuller information on the method used, write for a copy of "The Theory and Practice of Machine Control" to:

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S. 27

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Applied High Frequency is the modern heating technique now being adopted by more and more manufacturers for heat treatment of metals, and brazing and soldering operations.

It's the technique offering greatest economy because, it's faster, cleaner, saves material, and can be applied right in the production line, cutting out wasteful secondary handling.

Generators are simple to operate, and simple to change-over from one job to the next, and are therefore suitable for both continuous or small batch production.

The range comprises heating units with outputs of 1 kw., 7½ kw., 15 kw., 20 kw. and 25 kw. Write for a catalogue today.

Engineered by the Applied High Frequency Division of

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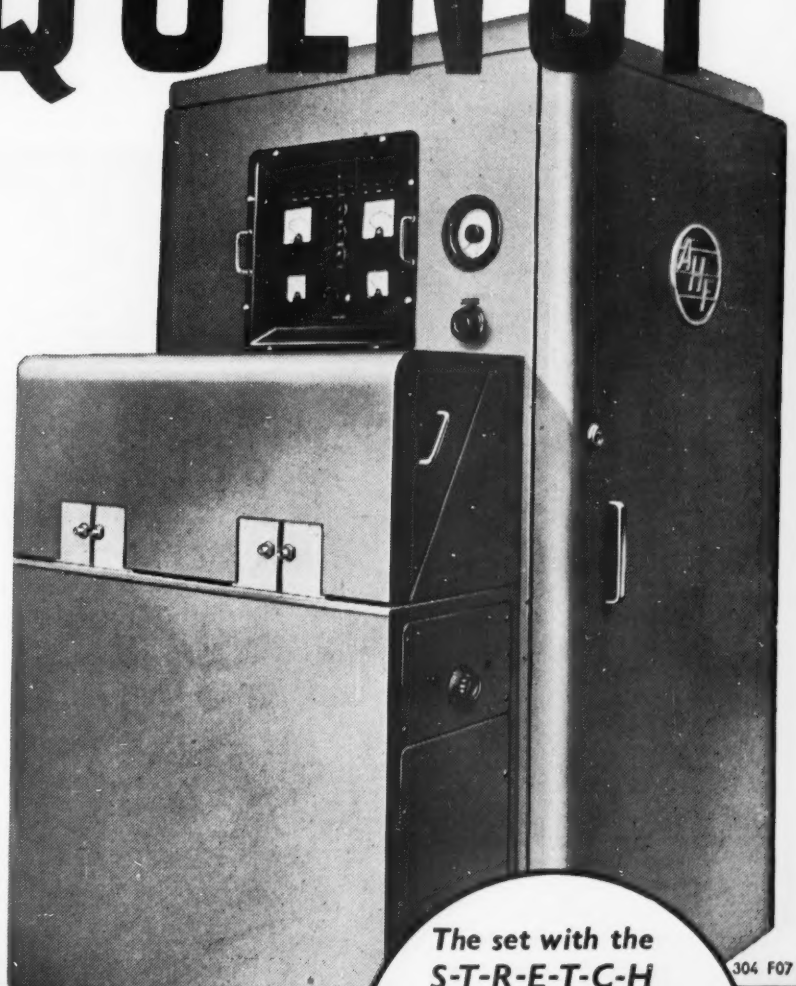
FREQUENCY

Today's cost-saving technique for Brazing, Soldering and Heat Treatment operations



304 F07

The set with the S-T-R-E-T-C-H

The 15 kw., 20 kw. and 25 kw. Generators are built in a common frame and are completely interchangeable due to the unit construction. Conversion from one output rating to another is quickly carried out in your own works by the addition of the necessary units.

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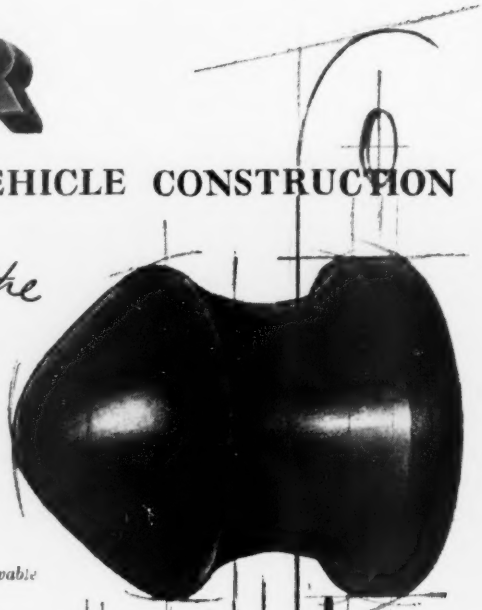
THE SILENT PARTNER IN VEHICLE CONSTRUCTION

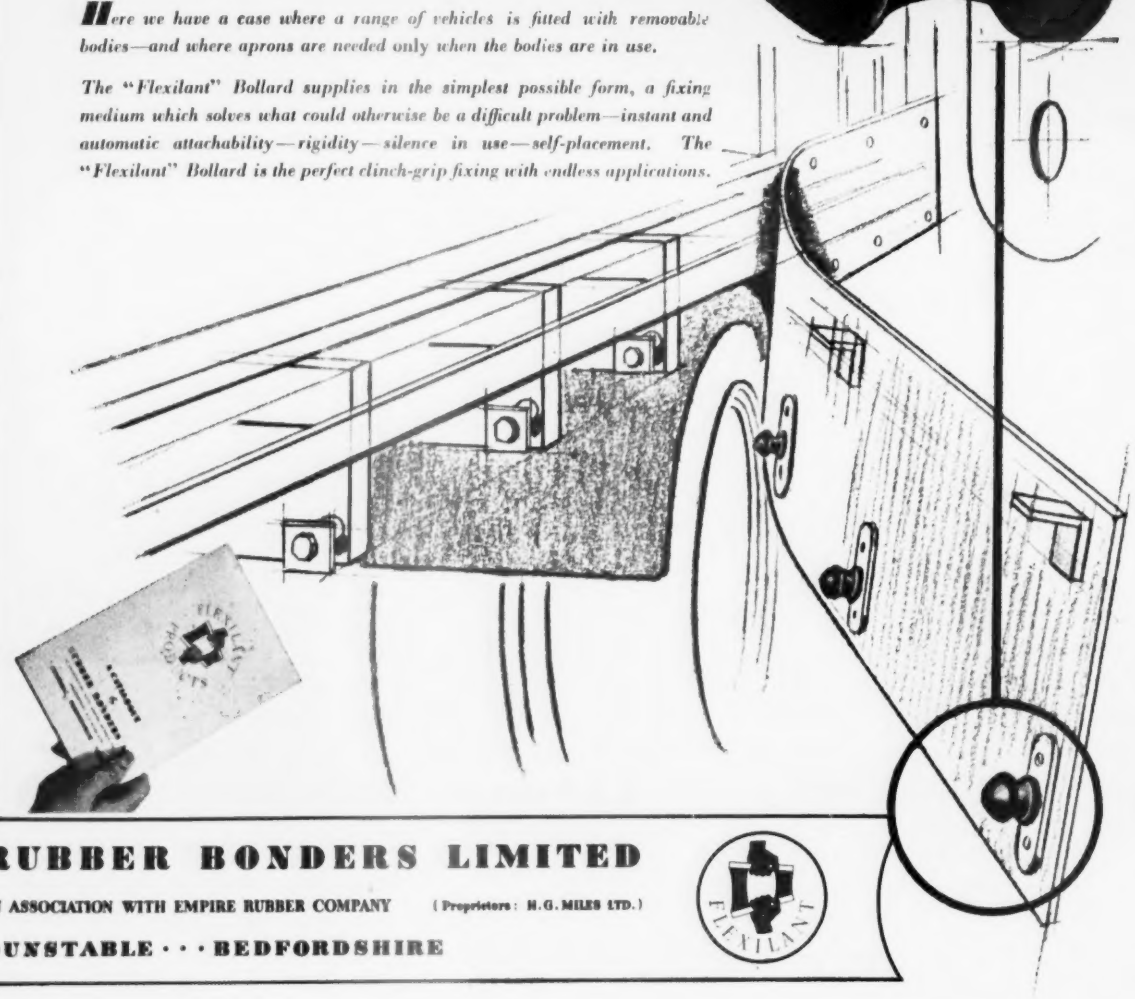
Another application of the

"FLEXILANT"
BOLLARD
SERIES 77



Here we have a case where a range of vehicles is fitted with removable bodies—and where aprons are needed only when the bodies are in use.

The "Flexilant" Bollard supplies in the simplest possible form, a fixing medium which solves what could otherwise be a difficult problem—instant and automatic attachability—rigidity—silence in use—self-placement. The "Flexilant" Bollard is the perfect clinch-grip fixing with endless applications.



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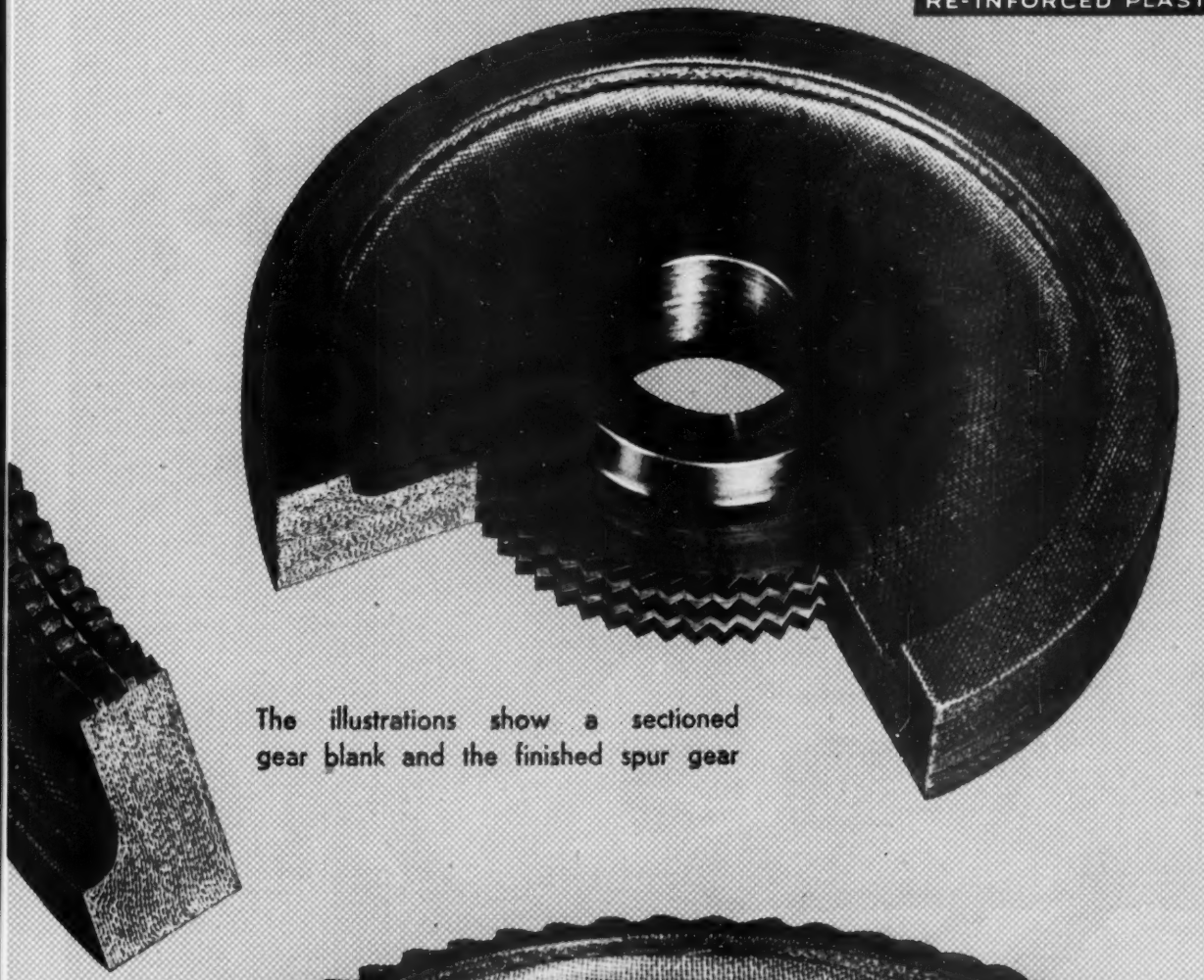
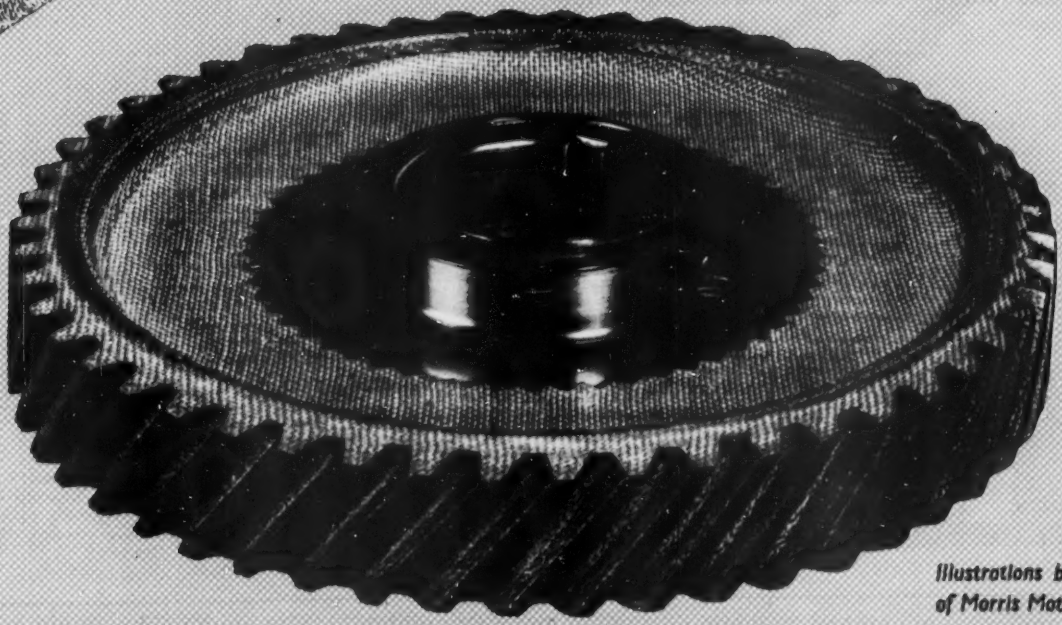


The illustrations show a sectioned gear blank and the finished spur gear



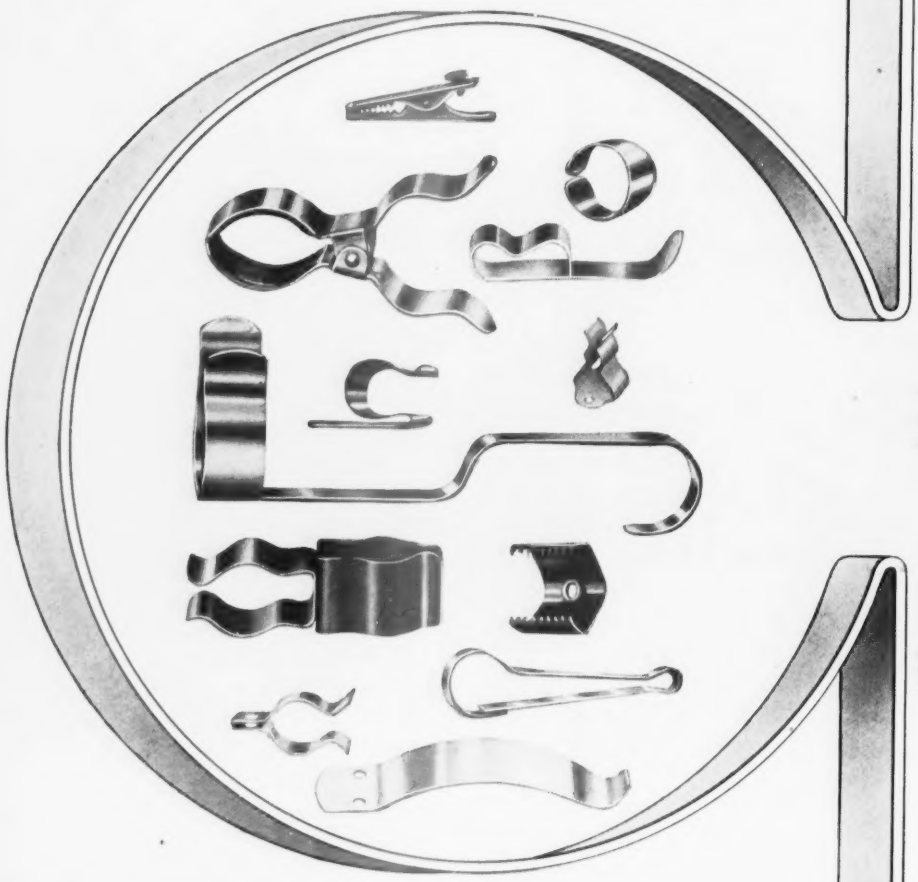
Illustrations by courtesy of Morris Motors Limited

THE BUSHING CO. LTD., HEBBURN-ON-TYNE

How are you fixed for CLIPS?

If you are having trouble with clips, bring your problem to us. We show here a few varieties, but these, mark you, are only a **very** small selection from our range of thousands of CLIPS . . . in every possible shape, size and quantity — in phosphor-bronze, brass, steel, stainless, plated . . . for every trade and profession.

And if you want a 'special' let TERRY'S Research Department design for you — after all, we've got 98 years' experience behind us.



and 5 very popular 'numbers'

80 and 81 — general utility clips — for tool racks, etc., from ¼" to 2" from stock.

300 — an exceptionally efficient drawing board clip, 5/- a doz. (inc. p.t.) from stock.

257 — a useful clip in black enamel, from ⅜" to 1½".

1364 — a clip for kitchen cabinets — rustproof finish.

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TERRY *Clips*

Sole Makers:

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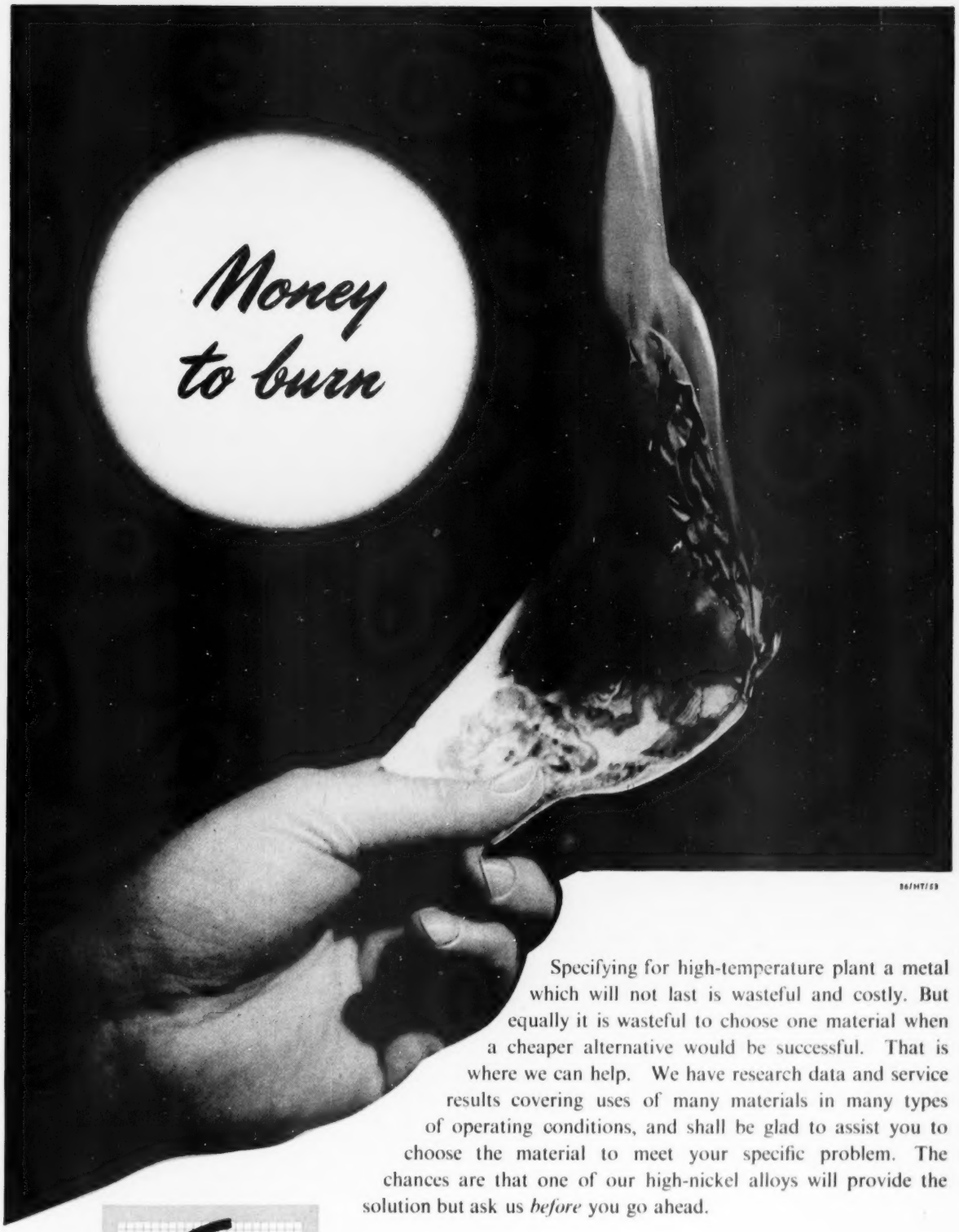

Specifying for high-temperature plant a metal which will not last is wasteful and costly. But equally it is wasteful to choose one material when a cheaper alternative would be successful. That is where we can help. We have research data and service results covering uses of many materials in many types of operating conditions, and shall be glad to assist you to choose the material to meet your specific problem. The chances are that one of our high-nickel alloys will provide the solution but ask us *before* you go ahead.

WIGGIN FOR COLD FACTS ON HOT METALS

HENRY WIGGIN AND COMPANY LIMITED · WIGGIN STREET · BIRMINGHAM · 16

FLYWHEELS
IN CAST IRON

by

DARCAST

Exhaustive tests at high speeds and production and use over a long period have proved the suitability of Darcast Flywheels in high duty cast iron for petrol and diesel engines and for compressors. Long specialised experience ensures homogeneity in the castings which is necessary to obtain uniform density with perfect balance, while the machining faces are free from hot-crazing and suitable for friction clutch operation. Our current production covers a wide range 7" - 24" diameter and 8 - 300 lb. weight in irons with a tensile strength of 15 - 24 tons per sq. in. Strict metallurgical control maintains the high quality of the material and advanced methods of mechanised production keep the cost of castings at the minimum. An interesting new leaflet is available on request.

DARTMOUTH AUTO CASTINGS LIMITED

SMETHWICK 40 BIRMINGHAM

Maximise my foot!

Sales? What are Sales? (said the Managing Director to this Economist Chap) I thought they were a kind of rite or orgy that Drapers held in the Spring. You say we might maximise our Sales by selling through Wholesale and Retail Channels? **Maximise!** What a word! Sir, we are concerned with Service, not with Sales.

We do not wish to sell our remarkable Power Tools over counters, like so much spaghetti. We wish to understand our Customer's problems and with him, to choose, and if needs be devise, the Tools to solve them.

Maximise our Sales! Sir, I find it a revolting and obnoxious phrase. Is there, then, merit in mere SIZE?

But the moment this Economist bloke had crept away the Managing Director slipped the Monthly Sales Chart from under his blotter. **"Another all-time record, Miss Buggins"** *he said complacently, the old twicer!*

Desoutter tools *put power into your hands*

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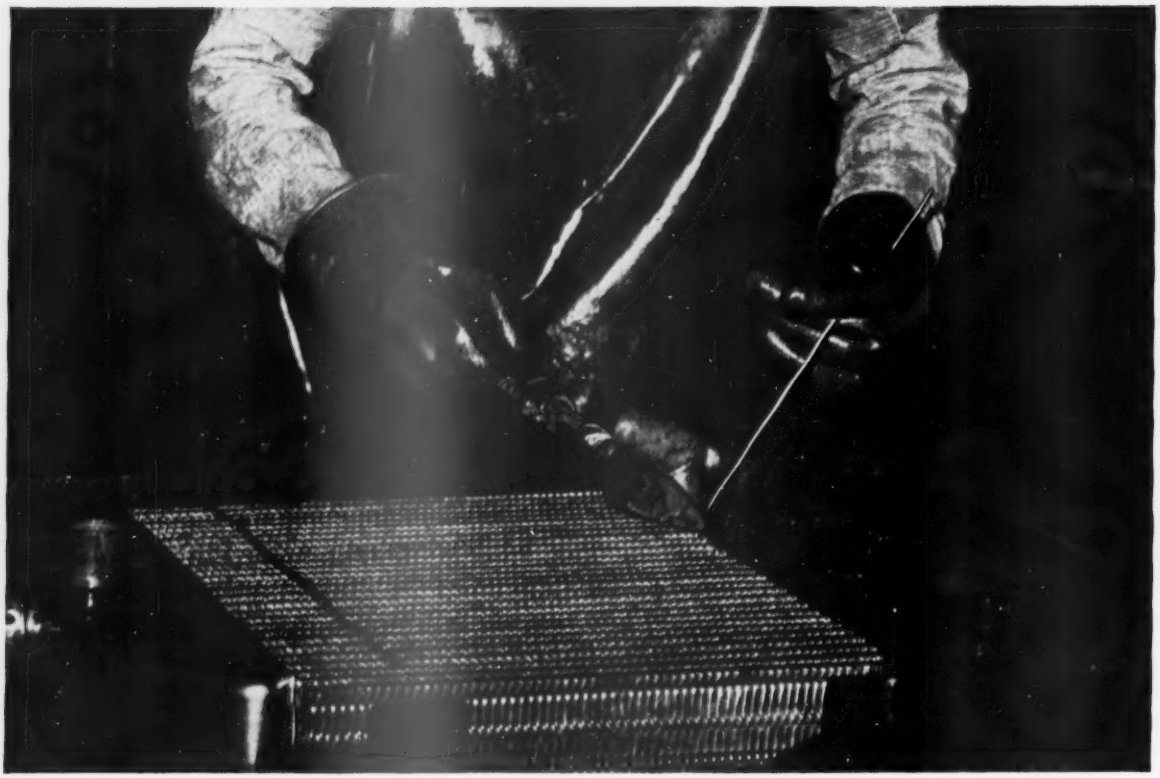


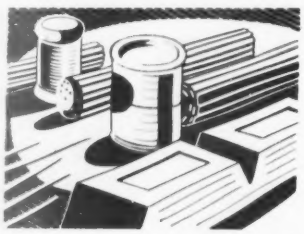
Photograph by courtesy of Vauxhall Motors Ltd.

Britain's *most modern* automobile plants

USE ENTHOVEN SOLDERS

The unique standards of purity and quality to be found in Enthoven Solders are an essential aid to the perfect operation of modern production technique. These standards are maintained by constant laboratory control during all stages of manufacture.

Enthoven Tinmans, Blowpipe, Ingot and Solid Wire Solders are supplied to B.S. Specifications 219-1949 and customers' special specifications.



Enthoven Body Solder is supplied in sticks to standard specifications or special alloys.

Entocene Solder Paint for pre-tinning on body work and for certain sweat soldering work is easy to use, clean and economical in operation. Our technical advisers and laboratory staff are available for consultation. Technical data, descriptive literature and samples gladly sent on request.

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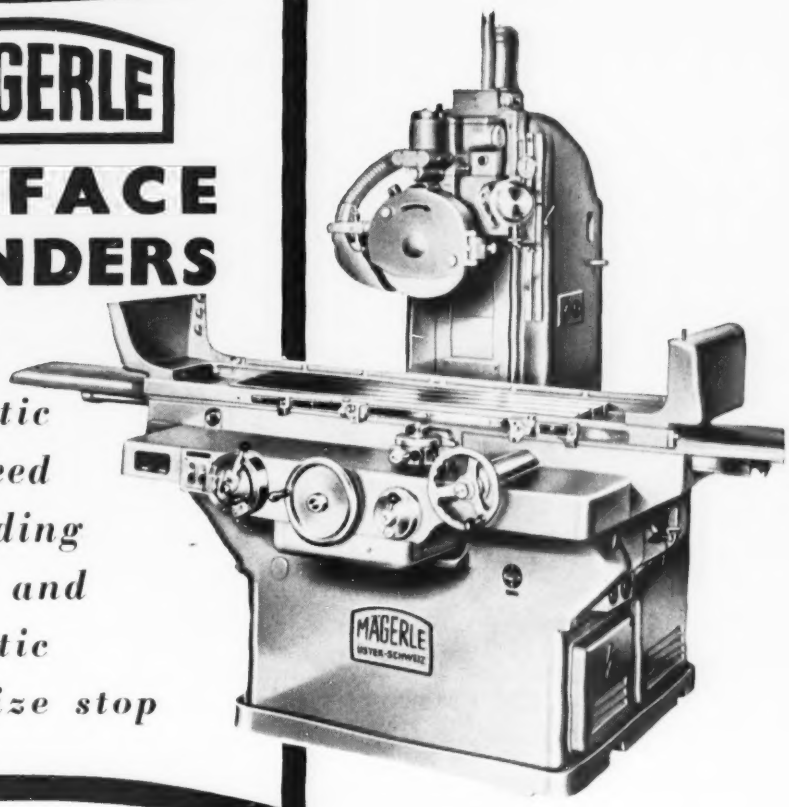
FOR UNUSUAL MACHINES •

MÄGERLE

SURFACE GRINDERS

With Automatic Down feed of Grinding Spindle and Automatic Final size stop



★ Automatic down feed of wheel head infinitely variable between 0·0001 in. and 0·0005 in.

★ Final size stop functions accurately to 0·0002 in.

★ Wheel diameter reduction by built-in diamond dresser is automatically compensated.

★ Hydraulic controls: Table speed infinitely variable 10-100 ft/min. Intermittent cross feed $\frac{1}{32} - \frac{3}{8}$ in. Continuous cross feed $\frac{1}{4} - 6$ ft/min.

★ Extremely simple operation by the most modern method of controls.

★ Efficient system of lubrication maintains maximum precision of table guideways.

★ Subdivision of drive into three separately motorised parts avoids complicated transmissions and lengthy piping.

Immediate Delivery

DEMONSTRATIONS AT OUR WORKS GLADLY ARRANGED

★ *Write for Catalogue*

THREE MODELS

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Table working surface (length) -	$28\frac{1}{32}$"	$39\frac{3}{8}$"	$47\frac{1}{4}$"
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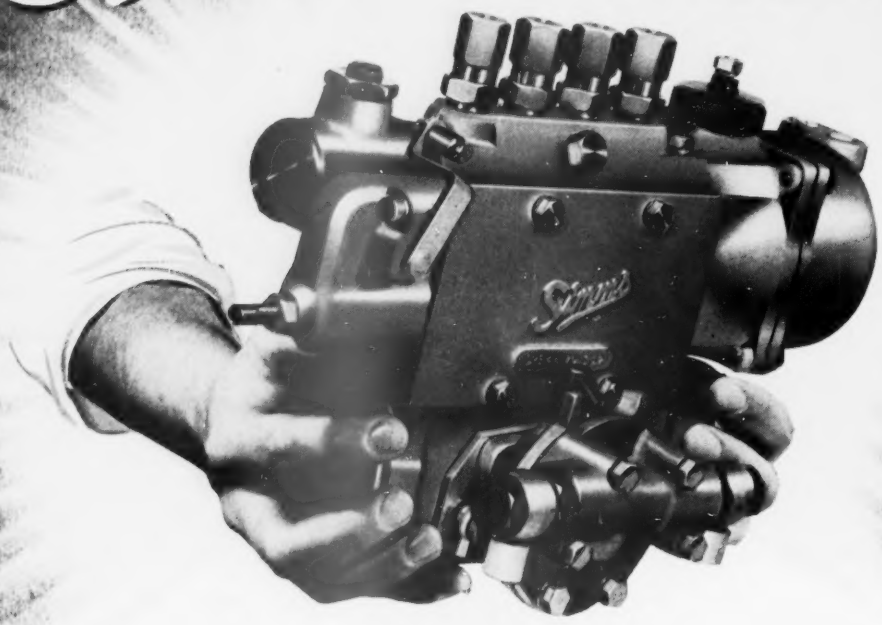
PHONE BATTERSEA 8888 (8 lines)

Simms PRESENT

THE NEW FUEL PUMP



Of Outstanding Design

Evolved primarily to meet the increasing stresses of smaller, high-speed direct-injection engines, this remarkably compact unit embodies—among other innovations—the new Simms G. P. pneumatic governor. Further information on this latest Simms product will gladly be supplied on request.

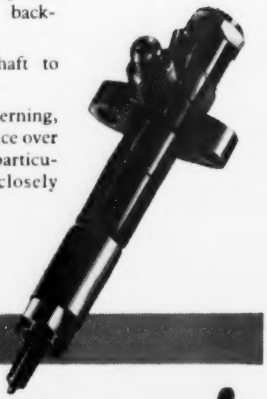
NEW... elements, highly resistant to wear from dirty or badly-filtered fuel.

NEW... method of adjustment simplifies calibration and reduces backlash.

NEW... large diameter stiff camshaft to withstand higher pressures.

NEW... method of pneumatic governing, giving consistant performance over the whole speed range with particularly steady idling and closely controlled run out.

NEWS



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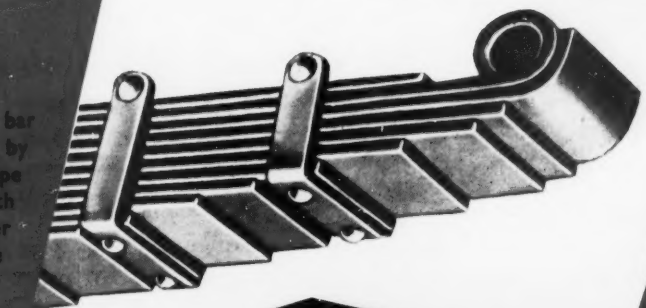
make them all

TORSION BARS

COIL SPRINGS

Coil, laminated and torsion bar springs are manufactured by Toledo Woodhead for every type of automotive vehicle — with shot blast treatment for longer fatigue life if required. We also supply Tine springs for cultivator and agricultural implement manufacture.



LAMINATED SPRINGS

We are also the manufacturers of KANTLINK SPRING WASHERS

TOLEDO WOODHEAD SPRINGS LIMITED · SHEFFIELD 3

TWS52

do it with 'dag' DISPERSIONS

CLEANER DIE FORGING & PRESSING

Treat top and bottom dies with an aqueous dispersion of **'dag'** colloidal graphite — the only lubricant effective at high metal-working temperatures — and you get these advantages :-

★ *Complete absence of fumes.*

★ *No harmful deposits.*

★ *No explosive forces set up, hence no cracking or pitting of dies.*

★ *Longer, trouble-free die-life.*

★ *Free flow of metal in both dies.*

★ *Perfect parting and the finest possible finish.*

In fact you have the most up-to-date and simple lubrication technique for all kinds of drop forging, press forging, hot stamping and other high temperature metal-forming processes.

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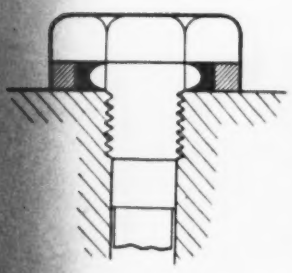
DISPERSIONS

TAS/AC33

BONDED DOWTY SEALS



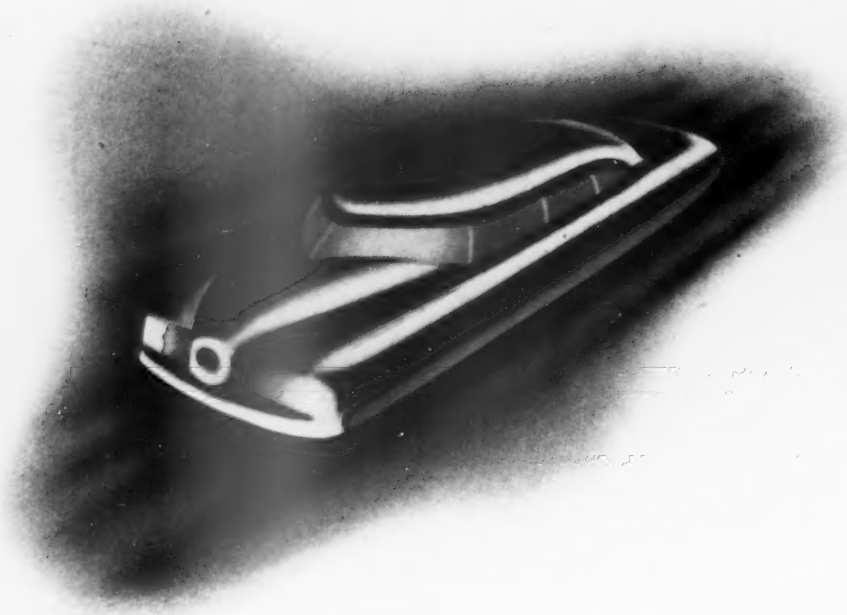
Especially in cases like this—

—where a washer must also act as a seal, a Dowty Bonded Seal will perform this dual role with maximum efficiency.
Handy for storage, Bonded Seals also provide simple fool-proof assembly and are suitable for use with either fluids or gases under a wide temperature range. Many of the standard sizes available are capable of with standing pressures up to 10,000 lb/sq. in.

B.I.F. (Birmingham)
STAND No. C.115.

By courtesy of The Ford Motor Co., Ltd.

DOWTY SEALS LIMITED · ASHCHURCH · GLOS.



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you will find

to be sure . . .

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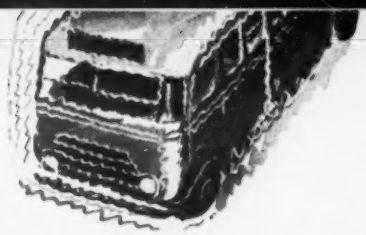
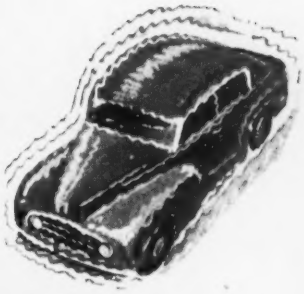
SMETHWICK DROP FORGINGS LTD. SDF SMETHWICK & KIDDERMINSTER, ENG.

Andre stops vibration



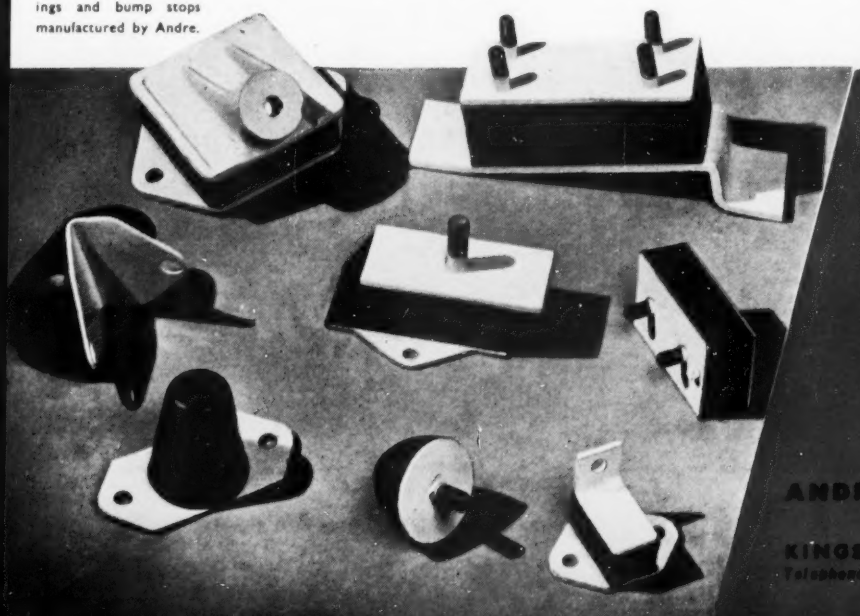


Andre RUBBER

Rubber-bonded-to-metal mountings provide the most effective and economic method of damping vibration. But the design must be right and the bond beyond question. You can be sure of both at Andre where the technical service and production facilities of a company specialising in bonding and moulding rubber are at your service. Our Technical Department welcomes the opportunity to co-operate in the development of new applications of engineering rubber.

Write for a copy of "Elastomeric Engineering Examples"—a photographic record of the wide range of rubber and rubber/metal components produced by ANDRE.

A selection of Rubber-bonded-to-metal mountings and bump stops manufactured by Andre.



ANDRE RUBBER

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WHICH ARE 40% LIGHTER THAN
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OUR ENGINEERS ARE READILY AVAILABLE FOR CONSULTATION ON NEW PRODUCTS

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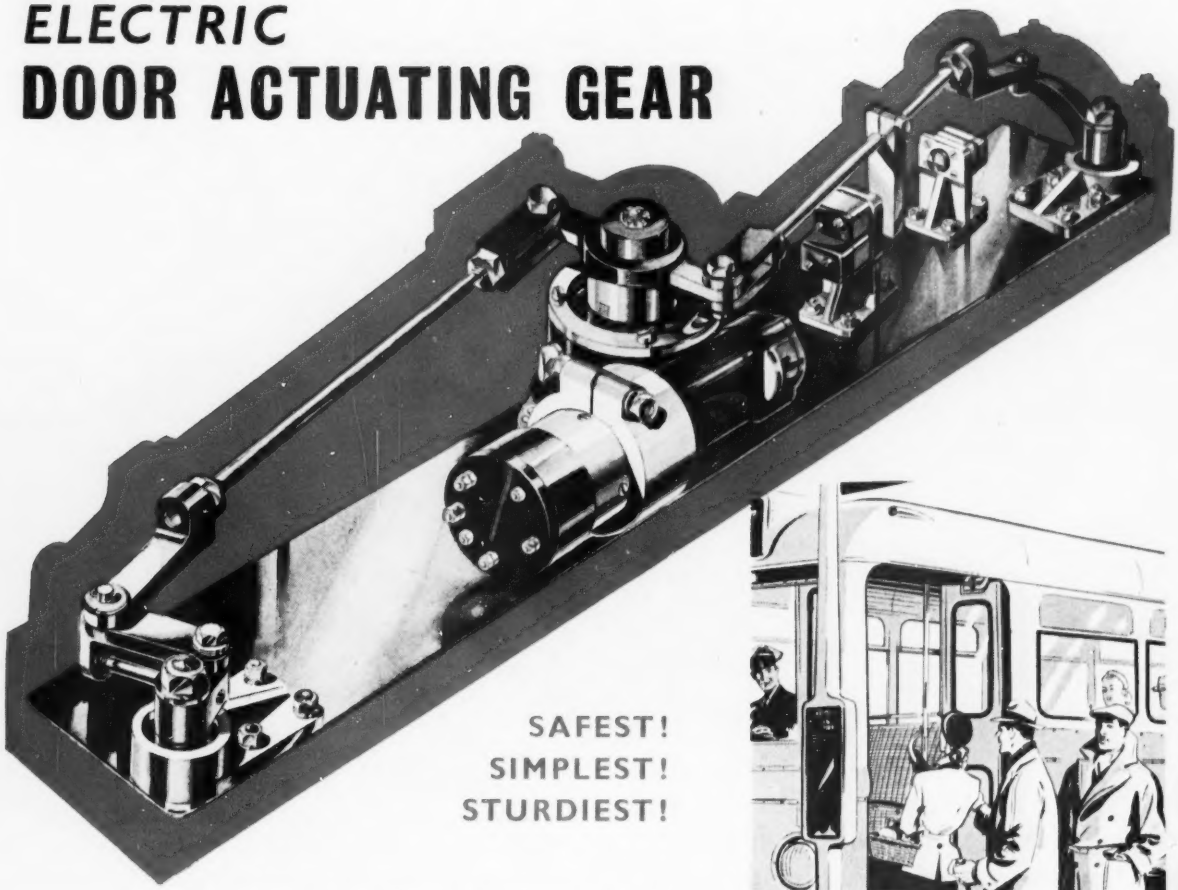

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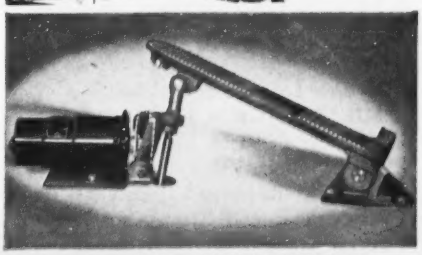

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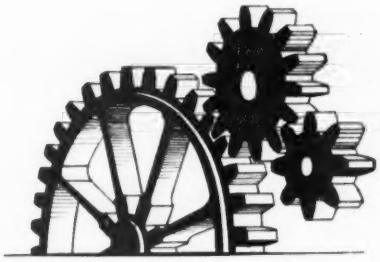

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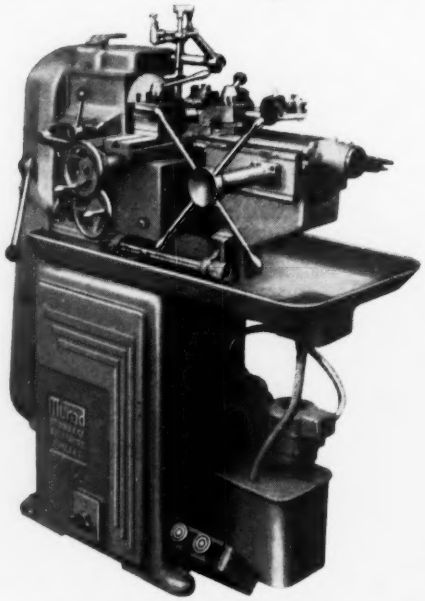

Murad

No1A *high speed*
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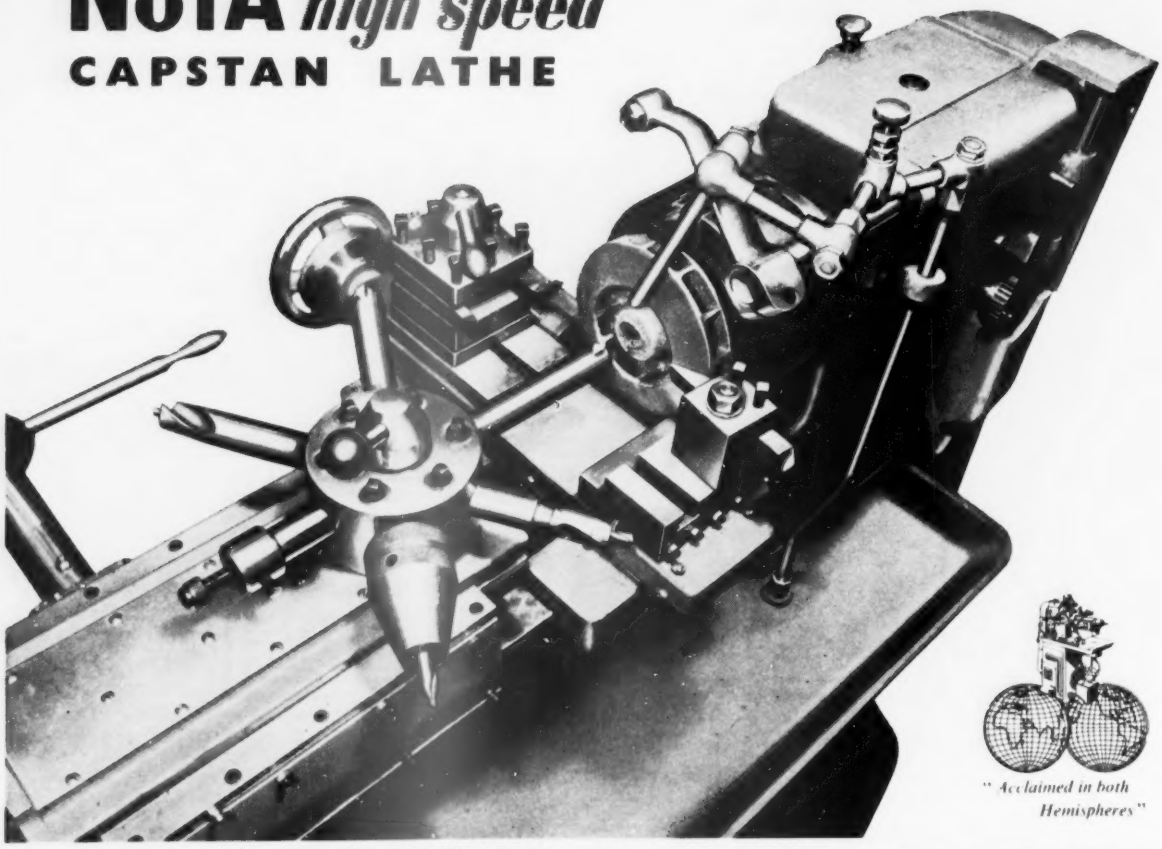

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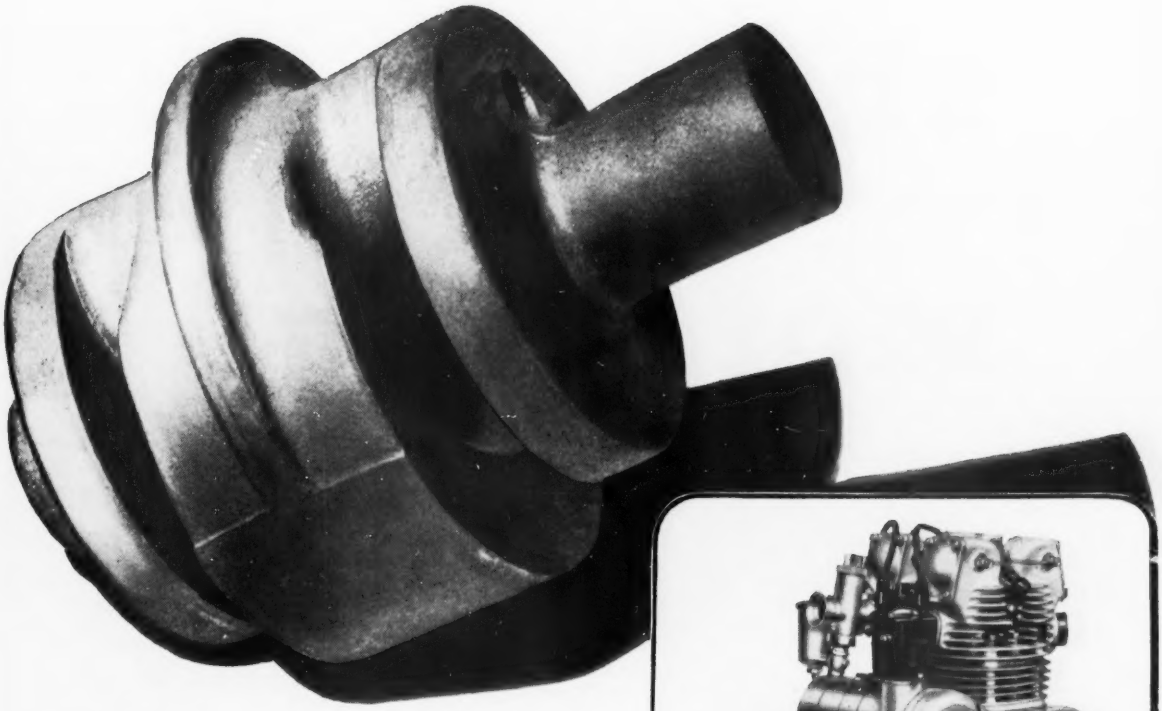

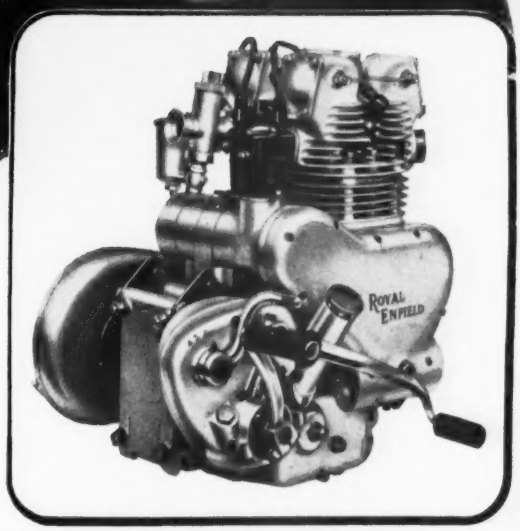

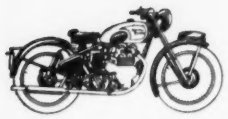

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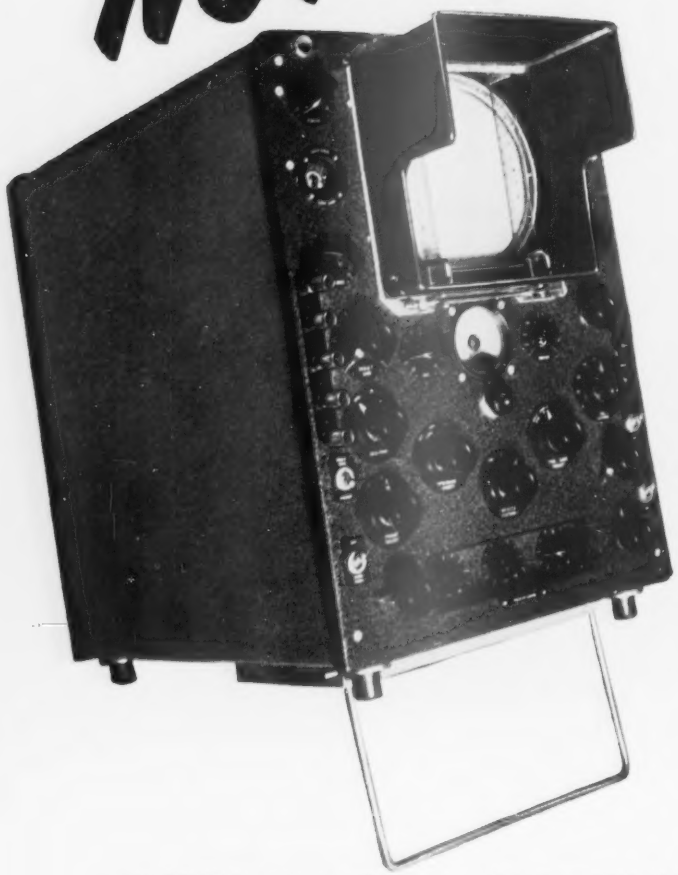

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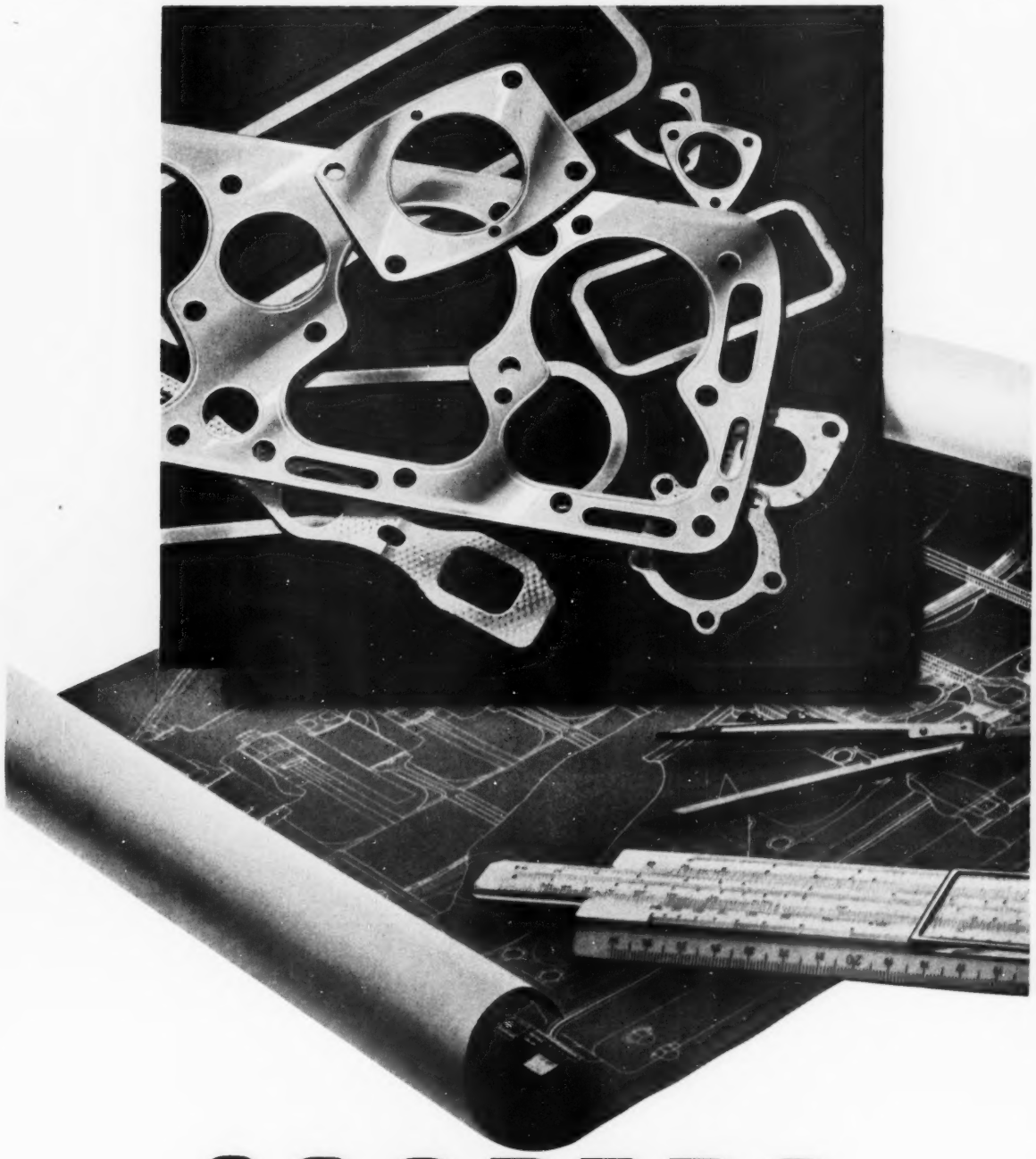

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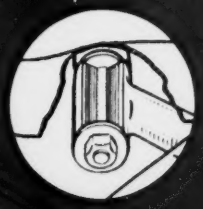
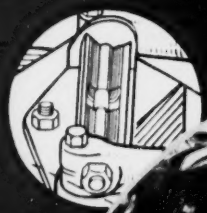
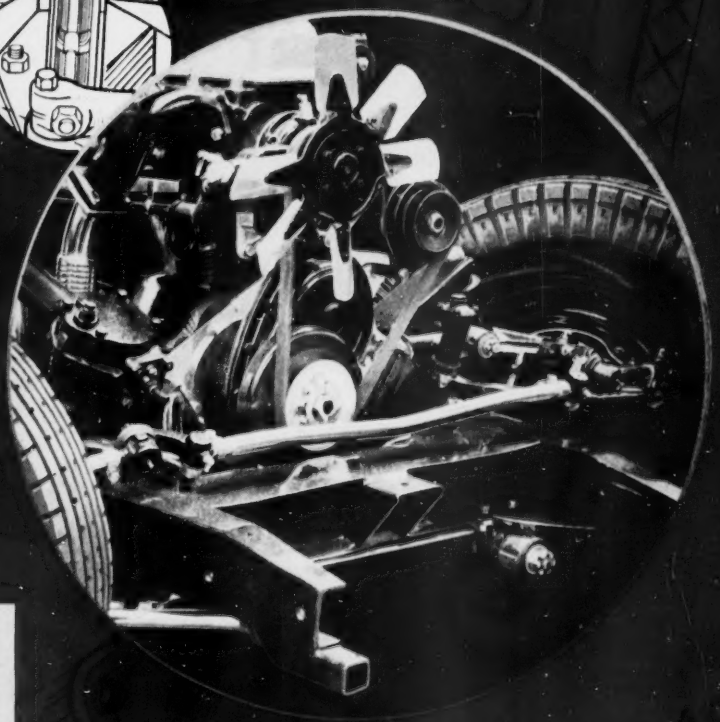
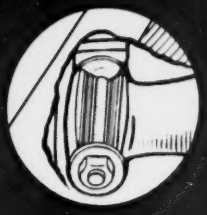






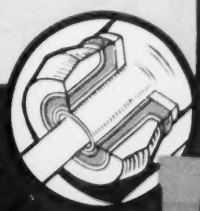


Automobile design to-day is in a healthy state of development: different designers show interesting differences in their detail design.

Here, for example, are three recent British designs of independent front suspension.

The SM 1500 (top) has Metalastik Ultra-duty bushes, all JAGUAR models (centre photo) have Metalastik Bonded-cone bushes, and the Jowett (bottom) has torsion-bar suspension with Metalastik axial snubber bushes having specially high resistance to end-thrust in addition to their high radial capacity.

It is significant that despite these differences, this lively state of development leads each designer to incorporate Metalastik units in order to attain the desired friction-free liveliness in suspension, and it is characteristic that we at Metalastik can meet the varying requirements from our extensive technical resources.



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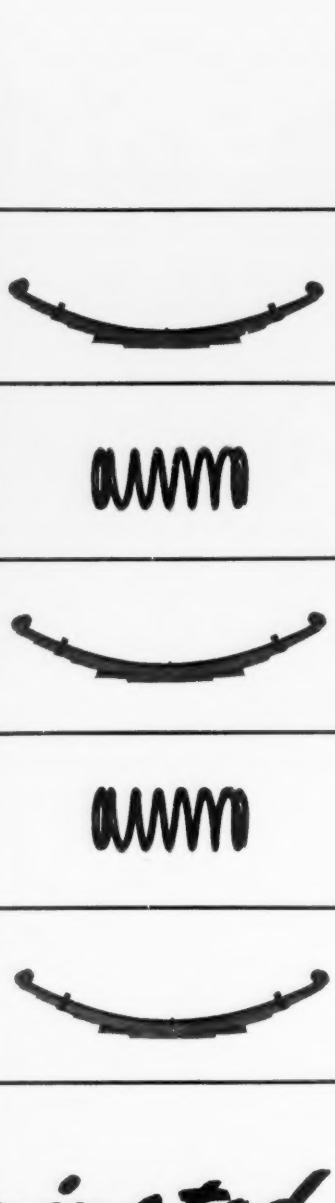

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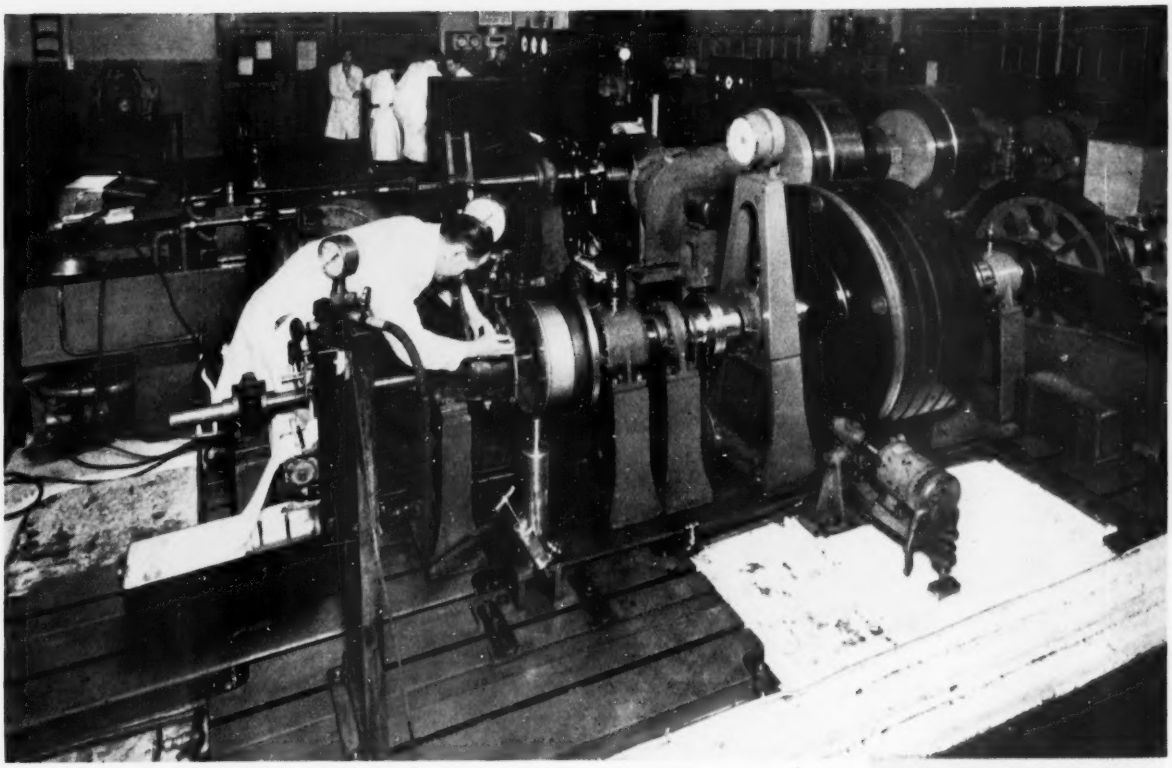

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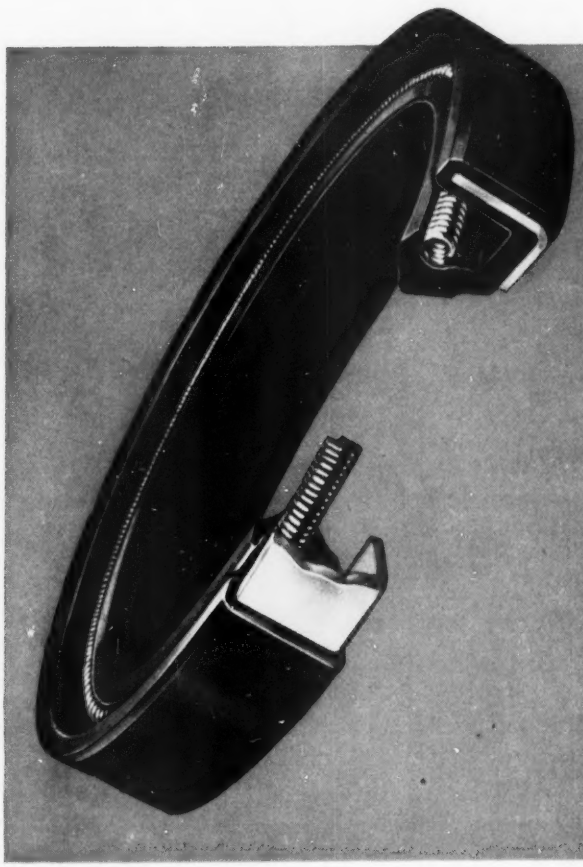

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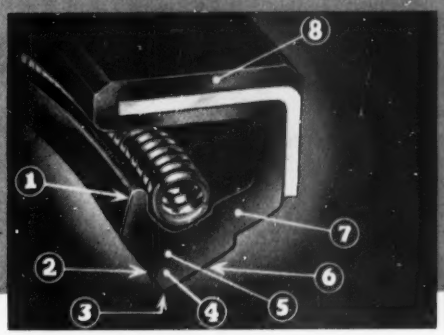

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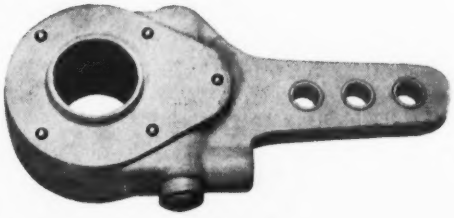

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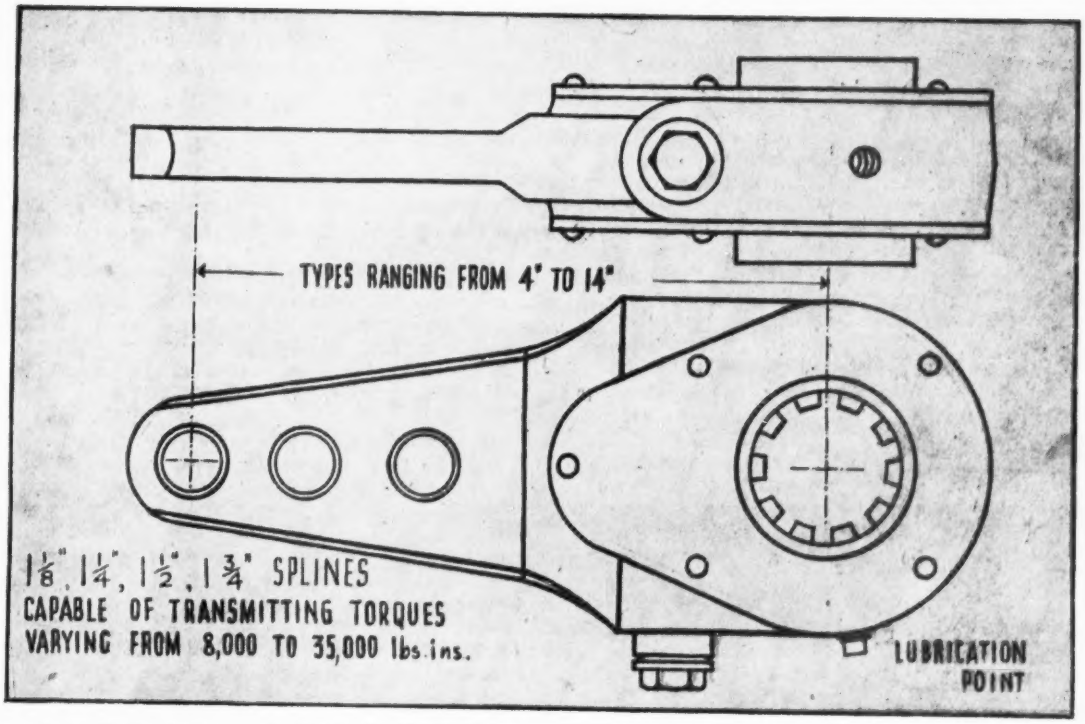
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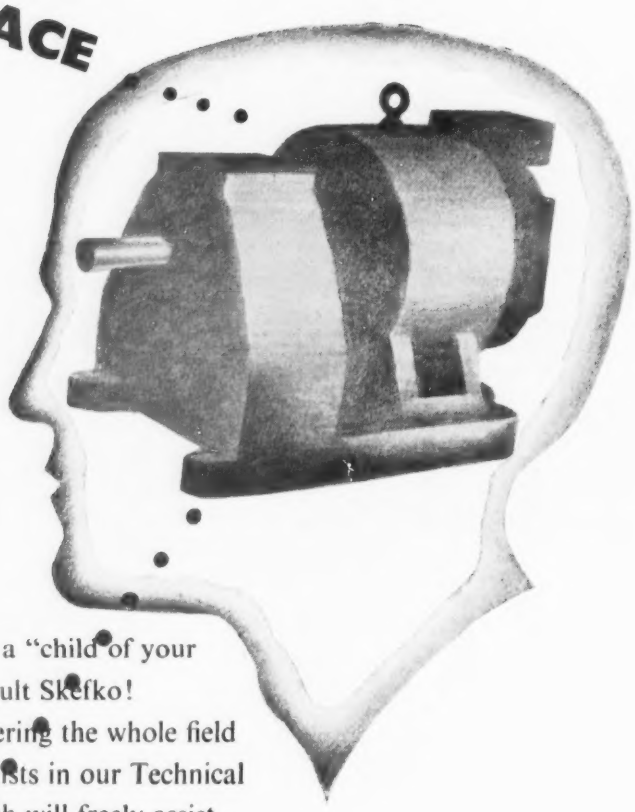

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Design, Materials, Production Methods, and Works Equipment

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Geneva

FOR the automobile engineer the various motor shows are primarily opportunities for making reasonably detailed examination of many different vehicles and for assessing possible trends for future development. The Geneva show, which is dealt with at some length elsewhere in this issue, certainly afforded the opportunity for comparisons, but the exhibits did not disclose any trends of marked importance, and particularly so far as chassis design is concerned. In general, the coachwork, rather than the mechanical details, offered most of interest.

The exhibit which without doubt attracted most engineering attention was the new Fiat 1100. This vehicle, which is now of chassisless construction, is likely to prove a formidable competitor in the popular light car market. Because of the interest the design evoked, it is worth while considering some design points in fair detail. The engine and front assembly are carried on a sub-frame, a method that has much to be said for it. To begin with, this method greatly facilitates the initial assembly of the group of components. It is also probable that major servicing operations can be more readily effected when the body can be lifted clear of a sub-frame assembly.

However, unless servicing requirements are carefully considered at the design stage, there is a possibility that minor jobs, such as the removal of the fuel pump or the changing of a filter, may prove very awkward, although there might be no difficulty in fitting these components during production. Here, it must be emphasized that these remarks are not intended to imply that the new Fiat 1100 will be difficult to service.

Despite the advantages of a sub-frame for carrying the engine and front suspension, it is doubtful whether this arrangement is the best possible from the structural point of view. The sub-frame is bolted to the structure, but practically all the other components are attached by welding, and it is generally unsatisfactory to employ both bolted and welded joints on one structure. Welds are comparatively unyielding in comparison with bolts, which generally have some clearance to be taken up. Consequently, in certain circumstances, the indiscriminate use of both types of joint may lead to practically the whole of the load being taken by the weld while the bolts deflect just sufficient to take up the clearance and are relatively lightly loaded. This could lead to structural inefficiency. As the new Fiat 1100 is some 2 cwt lighter than its predecessor, it may be conjectured that there has been a significant improvement in structural efficiency. It does not, however, follow that further improvement cannot be made.

An undesirable feature of this form of construction is that all the heaviest loads, namely, those from the engine and suspension, are concentrated on members of relatively small cross section. Because of this, the members must be constructed of heavy gauge material in order that they may be strong enough to carry the loads, yet they can scarcely be made as stiff as is desirable. We are, as yet, far from being convinced that a sub-frame is the best arrangement, but it may be that in a very small car no other layout is feasible because of space restrictions.

Nor is the arrangement without production disadvantages. In comparison with other types of chassisless construction, an additional operation is necessary to assemble the sub-frame to the body. At this extra operation 22 bolts must be fitted. Moreover, there is a distinct possibility that extra jigging is required.

An American development that does not, in our opinion, deserve to be followed, is the use of chromium plated cappings over pillars to hide open spot-welded joints. It is doubtful whether the saving in cost in comparison with the cost for lead-filled joints, can be of any real significance. Moreover, exposed chromium plating is not always as durable as might be desired. Furthermore, we must deplore the increased use of chromium plating for ornamentation.

One American two-door vehicle had a feature that is so advantageous and yet so simple as to cause surprise that it was not adopted earlier. This car has the front bench-type seat divided at one third of the width from the near side, and the near side portion is arranged to hinge forward. As a result, both the driver and one passenger can remain seated at the front while the rear passengers are free to alight. It is questionable whether a bench seat in a small car could be divided in this manner without an adverse effect on passenger comfort.

Specialist coachwork

It would appear that the Italian specialist coachbuilders, who were for so long regarded as the leaders on the Continent, are now having to yield pride of place to the leading Swiss firms. Nearly all the coachbuilders of both these countries have adopted what may best be termed modern styles, which may well provide leads for large scale manufacturers to follow, although with very definite modifications for reasons of production economy. The

specialists however, do not always display a very practical outlook and there were examples in which functional considerations had been sacrificed for the sake of that extremely arguable thing, a pleasant appearance. For example, headlamps were to be seen projecting further forward than the bumpers, there were windscreens with excessive rake and head-rails too close to the driver.

Commercial vehicles

A class of vehicle that is taking on increasing importance is the four-wheel drive, independently-sprung, cross-country type. The demand from any one section of users for this type of vehicle is probably hardly sufficient for manufacture in quantities large enough to allow the optimum production economies to be made. As such vehicles can be adapted to a wide variety of uses, including farming, lumbering, road work, quarrying, and general contractors work, there is no doubt that a worthwhile market could be created at home and in the under-developed territories.

The exhibits at the Geneva show and at other recent Continental exhibitions suggest that considerable development is going on in certain quarters. There is as yet, however, no sign that a really cheap four-wheeler is being developed, although the potential market for such a vehicle must be considerable. This is a field that should be well worth exploring. The best line of approach would appear to be that one, or more, of the large manufacturers should produce the chassis only and then sell to smaller organisations better adapted to the manufacture of the specialist bodies that would be required.

The British commercial vehicle industry has done remarkably well in export markets during the post-war years, but competition is now keener. If our position is to be maintained, every effort must be made to study the needs of operators. We in this country sometimes overlook the fact that in most countries commercial vehicles are operated over long routes with roads that are not nearly so good as the roads of this country. For export markets, manoeuvrability, acceleration, braking efficiency, high maximum speed and comfort are of prime importance.

More attention may have to be paid to these factors. Some organizations have a very good record in this respect, but others have not shown quite the same foresight. Greater attention to these factors would result in the provision of amenities that are not strictly necessary for operation in Britain, but it cannot be denied that they would be appreciated if the cost were not too high.

Management and Specialization

MODERN production techniques are developing along such lines that before long the question of new concepts of management will have to be considered seriously. The one definite trend in production to-day is that towards greater and greater specialization, and although we do not fully subscribe to the old aphorism about the specialist, we do think that modern conditions are leading to the development of engineers with an amazing fund of knowledge that extends only over a very circumscribed field. This, in itself is inevitable and is not to be deplored, but there is a danger that it may eventually lead to a paucity of men who are able to consider the problems of a large organization as a related whole and not merely as a number of separate parts.

As yet, this question has not assumed formidable proportions in the British automobile industry, but the possibility of its emergence in the not too distant future as a serious problem should be given serious consideration. Inevitably, the increasing complexity of industrial equipment and processes must call for the employment of more and more specialists, and this in its turn will increase the importance of administration and co-ordination.

Therefore, the engineer with a wide general knowledge will also have an important part to play. Only such a man will be able to co-ordinate the work of specialists in many different fields. There is, of course, no one reservoir from which such men can be drawn, but it would appear that the main source of supply should be from university-trained graduates who have not specialized too narrowly during the period of academic training.

It does seem that in view of present-day trends in industry, and perhaps more particularly those of the automobile industry, the question of technical education requires serious consideration. Tentatively, it would appear that pure technology should be left chiefly to technical colleges, although many of the existing colleges will have to make considerable improvements in their existing standards, while the universities should remain true to their name and provide wider and less narrowly specialized courses. This problem of co-ordinating the work of specialists is already causing some degree of concern in the United States of America, and there is little doubt that except in the field of pure science that country is ahead of Britain; there is now need to consider the question in this country.

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THE GENEVA SHOW

Some of the New European and American Exhibits

As usual the annual Geneva motor show was of interest from every aspect of automobile engineering. This was because of its international character. French, German and Italian manufacturers of both commercial vehicles and private cars are all keen to exhibit, since part of the frontiers of each of these countries is common with that of Switzerland, and it is relatively easy for visitors from all three countries to go to the show.

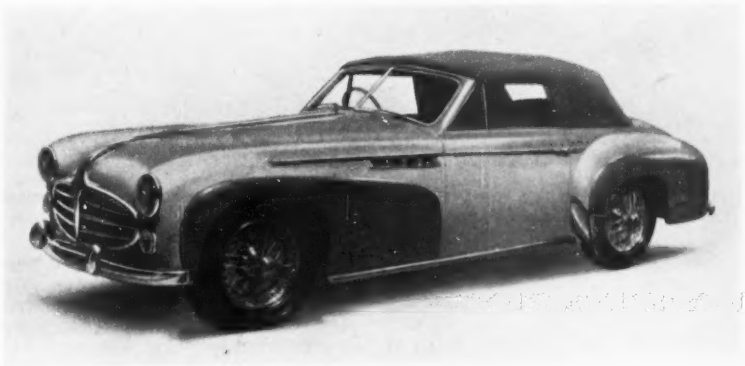
A further inducement is the fact that Swiss francs are a hard currency. This is an incentive to British as well as to foreign manufacturers to display their products in Geneva. It remains to be seen whether competition from European producers will increase as their home markets become saturated and they have to turn elsewhere to sell their vehicles.

In one respect, the Geneva show was somewhat disappointing. Some of the German exhibitors, not unnaturally, withheld their latest developments in order to introduce them for the first time at their own show, which opened on the 19th March at Frankfurt. Nevertheless, there were some interesting features to be seen.

American manufacturers exhibited their 1953 models, which have already been displayed in the United States. There are no restrictions on the sale of American cars in Switzerland; this is an incentive for them to take their latest cars to the Geneva Salon. Unfortunately, there is not the same inducement for them to display at the London Show.

Trends in styling

Among the Italian and Swiss coachbuilt products, marked trends could be distinguished in body styling, except where radiator grille arrangement was concerned. On this subject, opinions seemed to be divided. Some coachbuilders have adopted the mouth organ type of opening, while others incorporate an almost rectangular opening with the sides tapering in slightly towards the bottom and with a slight upward slope of the top edge from each side to the centre. In most cases, where nearly rectangular openings are employed, vertical chromium plated bars of small cross section are used for the grille, which is generally set about an inch in from the edges of the opening.



On this Delahaye, great pains have been taken to avoid a slab sided effect

In almost all models, fairly plain and serviceable bumpers are employed together with overriders. A few examples were seen of the buffer type of bumper in the form of vertical chromium plated projections at the front and rear respectively of the front and back wings. It is, however, difficult to see how these can afford much protection to body panels, and it is logical to assume that they will not have a very widespread appeal. Most of the vehicles have some kind of chromium plated moulding positioned on the longitudinal centre line of the bonnet to break up the otherwise plain appearance of the bonnet top.

Practically all have vertical fin-like extensions of the rear wings. If the aerodynamicists' claims for the effectiveness of this arrangement in giving stability in high winds are justified, it seems likely to continue to be popular, particularly in high performance vehicles. Another device of which many examples were to be seen is the large horizontal chromium plated feature fitted near the lower edge of the boot lid for use as a handle with which to lift it. From this it follows, of course, that the lids are generally hinged at the top. A number incorporate external chromium plated hinges to relieve the plainness of the rear decking. This is particularly desirable in some convertibles.

Full width rear windows were fitted almost exclusively, and most are extended round the quarter panels so that the canopy at each side is reduced to what is virtually a rear pillar. These windows look best when set at a fairly large angle from the vertical; some that are not arranged in this way look particularly ugly. This is probably because an exaggerated slope is necessary to offset the effect, created by the transparency of the glass, of a discontinuity of line between the canopy and the boot profiles. Another difficulty with the large rear window arrangement is that of avoiding discontinuity of line at the rear pillars.

So far as the sides are concerned, almost all models incorporate a horizontal chromium plated strip extending most of the length of the car between the wheels. On some cars this strip is about 12 in to 16 in above the sill; on others it is on the top edge of the sill where it serves to protect the paintwork. The principal object in fitting these strips is to break up the otherwise plain areas of the slab sided

Chromium strips, either along the top of the sill as in this Graber Alvis or in a higher position, are used to break up the plainness of the body sides

The Plymouth Belvedere is an example of one of the lighter and more pleasing American front end styles

bodies which are so popular because they give maximum interior width. Disc wheels are fitted almost universally, and there are few, if any, examples of wings extended down to hub level either at the front or at the rear.

Among the American exhibits, it was more difficult to detect positive trends in styling. Bumpers generally are slightly less massive than hitherto, but they are still appreciably larger than those on European cars. The Mercury is an example of the employment of a chromium plated skirt swept inwards below the front bumper. The overriders are pressed in this panel, and immediately above them forward extended bullet-shaped caps are mounted on the bumper. This low placing of the larger chromium plated components has the effect of appreciably lightening the front end appearance.

Radiator openings are mostly low and horizontal with chromium plated bars. In many cases the head lamps are well recessed behind their bezels. The wide bonnet tops are decorated with multiple, or wide chromium plated strips on the longitudinal centre line of the lid. Alternatively shorter and heavier mouldings surmounted by rocket and missile type mascots are employed for the purpose.

Presumably it is the need to break up the large plain expanses of sheet metal on the big American cars that has led to the adoption of such a multiplicity of chromium plated ornaments. Not only are these now used to accentuate the wing lines but they are also applied at the back end to almost as great an extent as at the front. Large bezels for the rear lamps in many cases cover the whole of the rear end of the fin-like wings; equally large boot lid handles, also chromium plated, are fitted; motifs of various kinds add to the confusion.

The most noteworthy changes in American design have been above the waist line. Roof canopies are now of the well-known "hard top" shape, with the rear windows extending round the rear quarters. Although the rear passengers do not generally benefit by being able to see out of the rear windows easily on the side on which they sit, by only a slight turn of the head a good field of vision may be obtained on the opposite side. Convertible models appear to be increasing in popularity.

Windscreens in general are very curved, and consequently the front pillars are closer to the driver than hitherto. This, coupled with the fact that the pillars are often relatively wide (in one American model they are over 4½ in wide), gives rise to a fairly large blind angle in the driver's field of vision. The tendency to use large section pillars might be accounted for by their use as a structural member in chassisless vehicles. Despite the fact that this practice is widespread, it is felt that the strength of the structure should be concentrated in the lower portions of the body and the sill. Not only does this avoid undesirable restriction of the range of vision, but it enables changes to be made more readily to convert a saloon to a drop head version.

There is a marked tendency to use chromium above the waist, and not only for covering the rubber glazing strip. Chromium drip channels are fitted at the cant rails, and external cappings are mounted on windscreen pillars, as well as on the pillars between the side and rear windows. In some cases this device has been adopted to cover open welded joints and so save the cost of soldering.

There were several other interesting features to be seen among the American cars. The two-door Studebaker Champion has a heavily framed rear quarter light which from both inside and out looks rather like an almost rectangular porthole of the type seen in ocean liners. The panel between this and the rear light is about 8 in

A long sweeping line pressed into the body panels breaks up what would otherwise be the plain appearance of the slab side of the Hudson Jet

The Ford Crestline Sunliner is an example of the use of chromium plate to elongate and accentuate wing lines

wide, measured horizontally. An open spot welded joint extending the full width between the canopy and the rear quarter panel is plainly visible. To improve forward visibility, a very low bonnet is employed. This is done by placing the V8 engine behind the front wheel centres and having an unusually wide radiator block. This makes the front overhang appreciably less than that at the rear. Like most of the other models from across the Atlantic, this car has press-button type door handles.

Another feature of interest seen on several other two-door cars concerns the front seats. The squab is split vertically about one third of the width from the nearside so that two passengers can remain seated in the front while the remainder of the squab is hinged forward to allow other passengers to enter the back seats. The other portion of the squab can also be hinged forward so that access to the rear seats can still be gained from the driver's side.

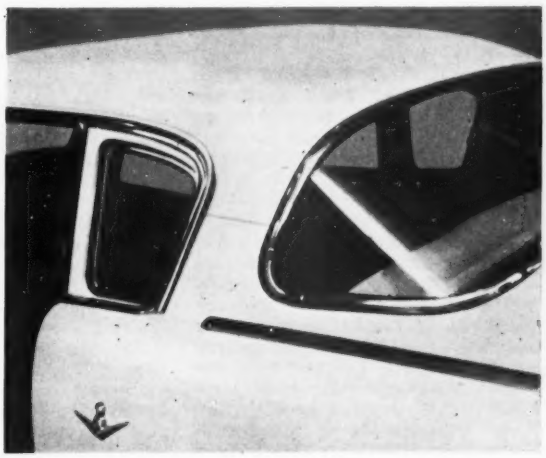
On the Studebaker Commander, unusually large headlamp bezels are employed and, because of the low bonnet line, a somewhat unconventional grille has been adopted

The Fiat 1100

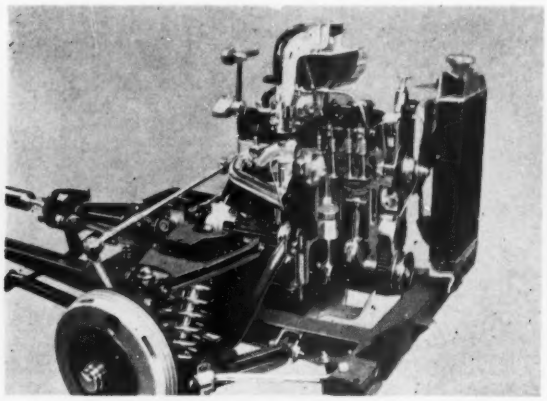
Undoubtedly one of the most interesting exhibits in the car section was the new Fiat 1100. This is a four-door saloon designed to seat four people in comfort. It can be obtained in Switzerland for 7,540 francs in the de luxe form or for 6,950 francs in the standard form. The principal overall dimensions are: length 12 ft 5 in, width 4 ft 9½ in, height 4 ft. 10¼ in, turning circle 34 ft 4 in. At the front the track is 4 ft 0½ in and at the rear 3 ft 11¾ in, and the wheelbase is 7 ft 8 in. The dry weight is 1,755 lb and the kerb weight is 1,800 lb. This is about 220 lb lighter than the previous model. The tyres are 5.20 × 14 in, and a 32 litre petrol tank is fitted.

The engine is the well-proved four cylinder 1,089 c.c. unit in which few modifications have been incorporated. Its bore and stroke are respectively 68 mm and 75 mm, and the compression ratio is 6·7 : 1. At 4,400 r.p.m. the b.h.p. developed is 36, which represents 33·1 b.h.p./litre, or 45 b.h.p./ton (kerb weight). Maximum torque is developed at 2,200 r.p.m. and is 50·6 lb-ft.

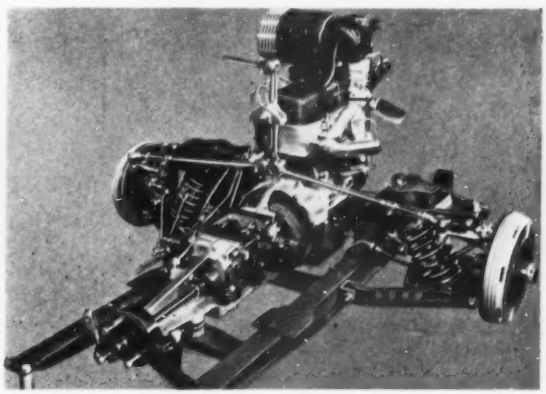
The suspension of this power unit is somewhat unusual. Two large sandwich type units are employed at the front, one on each side. Although the usual vee arrangement has been adopted, the units are in a transverse plane which appears to be about an inch behind the centre of gravity of the unit. Presumably this position has been chosen so that the mountings may be bolted to the cantilever brackets carrying the coil spring and shock absorbers of the front suspension. Another unusual feature of these mountings is that each rubber sandwich has a hole through it in which is housed a compression coil spring in a guide tube to provide the necessary vertical reaction to the weight of the engine while remaining relatively flexible torsionally. Metal brackets are positioned just clear of the front and rear faces of the rubber to provide fore and aft location. No engine tie is employed. At the rear, the unit is carried on a single coil spring which is on a cross member beneath the gearbox rear extension. Coaxial with the spring is a tie bolt carried in the rear extension and fitted



An unusual rear quarter light arrangement has been adopted on the Studebaker Champion Regal



A conical rotor is employed in the water pump of the Fiat 1100, and an oil strainer is incorporated in the pressure line from the pump to the relief valve



The engine and front suspension are mounted on a sub-frame, which is bolted up to the chassisless structure of the Fiat 1100

In the Fiat 1100, a strut behind the dash supports the bracing member for the front of the sub-frame

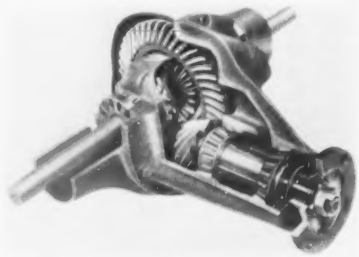
Lightening holes are used extensively in the cant rail of the Fiat 1100

at its lower end with a rubber rebound stop which bears against the underside of the cross member.

A conventional single dry plate clutch is employed; its bellhousing is bolted on to the cast aluminium gearbox casing. There are four forward speeds and one reverse. Constant mesh second and third speed gears are employed, and synchromesh operation is obtained on all except first and reverse gears. The gear shift control is on the steering column. The overall gear ratios are: top 1 : 1, third 1·57 : 1, second 2·38 : 1, first and reverse 3·86 : 1.

Immediately behind the gearbox rear extension is a transmission brake operated by the hand brake lever which is mounted on the floor by the tunnel. To the rear of the transmission brake is a flexible coupling, and integral with its rear spider is the sleeve portion of the sliding joint. A conventional cardan joint is used at the other end of the propeller shaft. The semi-floating hypoid bevel rear axle unit is housed in a banjo-type casing; the axle ratio is 4·3 : 1.

Leaf springs, underslung below the axle, are employed in the rear suspension. Grease nipples are provided for lubrication of the spring eyes and shackle bearings. Telescopic shock absorbers are attached to the lower plate for the U-bolts securing the springs. An anti-roll bar is fitted on the front as well as on the rear suspension. At the rear, brackets welded to the axle tubes inboard of the springs carry the rubber bushed drop links for the anti-roll bar. Carrying the centre



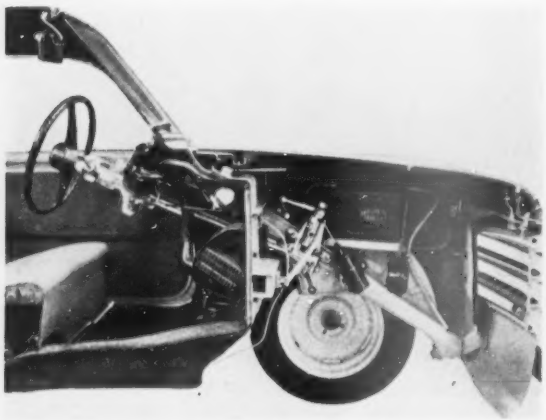
The rear axle of the Fiat 1100

portion of the anti-roll bar are two D-shaped rubber bushes, split horizontally, and fitting in bearing caps. They are bolted up to a body cross member at the rear of the seat pan.

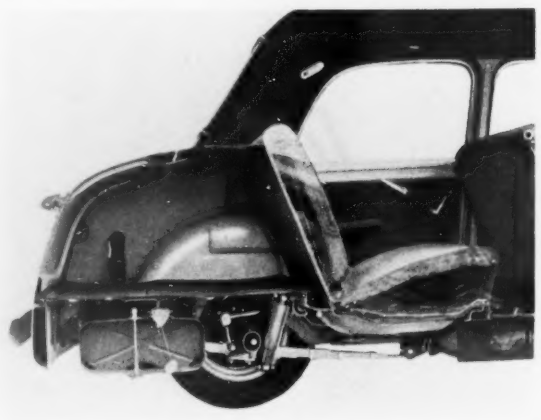
The front suspension, together with the engine, is mounted on a sub-frame. This is a most convenient arrangement from the point of view of both the sub-assembly and the final assembly into the vehicle. From each side member, a cantilever box section bracket extends upwards and outwards to support the top end of the coaxial coil spring and telescopic shock absorber. It also supports the single pin on the ends of which are pivoted the two arms of the upper wishbone. This pin is secured to the cantilever by two bolts. These bolts are each passed diametrically through the pin, a sleeve which takes the axial thrust from the inner end of the adjacent bearing, and through the flanges of the cantilever member.

Both the upper and lower wishbone arms are of pressed steel channel sections with their flanges facing outwards. Lightening holes are incorporated in the vertical walls of the lower arms only. The inner bearings of the lower wishbone arms are carried in heavy U-section lugs, each of which is secured by a single bolt passing transversely through the side member. Camber angle adjustment is effected on assembly by the insertion of shims between these lugs and the side member.

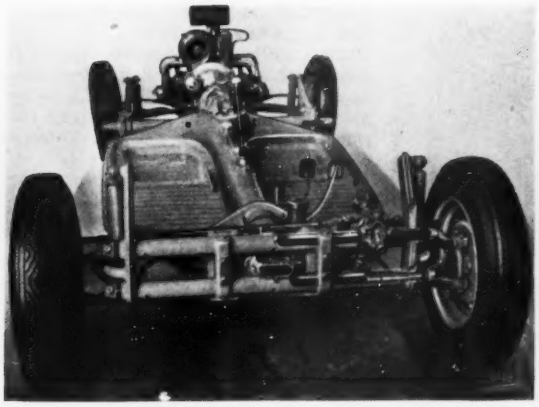
Each of the outer wishbone bearings, both at the top and bottom, is carried on a small trunnion forging on the swivel pins. These pins are well spaced and are integral with the stub axle forging. All bearings are grease lubricated. The brake drum rims are finned, and ducts are incorporated in



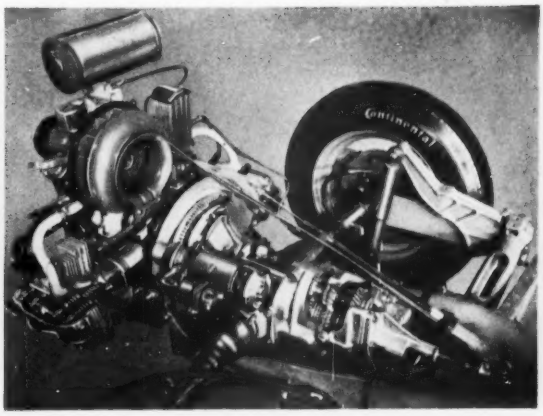
The steering box is mounted high up on the bracing member for the front of the sub-frame



A cross member formed at the junction between the boot floor and rear seat pan carries the shock absorbers and anti-roll bar



Laminated torsion bars are used on the front suspension of the Volkswagen



A synchromesh gearbox is now incorporated in the rear-engined Volkswagen

the vertical walls for cooling purposes.

At the front the ends of the anti-roll bar are carried in bushes mounted directly on the lower wishbone arm, and the more usual drop links are not employed. Axial movement in the rubber bush and flexing of the torsion bar arm accommodate the differential motion between the bar and the wishbone arm. This is possible because of the relatively acute angle between the two components. Bushes similar to those at the rear carry the centre portion of the anti-roll bar. They are bolted up to the gusset plate underneath each side member at its junction with the front cross member.

A somewhat unusual steering layout has been adopted. The trailing steering arms are connected to each end of a divided track rod. Above the clutch housing, the two pieces of this rod are carried on the lower end of a vertical, centrally pivoted lever. The upper end of this lever is connected to a horizontal rod to the arm on the cam and roller type steering box. This box is mounted on a bracing strut between the front of the frame and the dash, while the centrally pivoted lever is carried on a cross member between the bracing strut on each side.

The structure can be described as being a cross between the unitary and chassisless types. It has not a continuous frame welded under the floor in unit with the body, but incorporated in the body are a number of cross members, as well as the two bracing members which support the front end of the sub-frame carrying the engine and suspension. That the structural efficiency is high would appear to be demonstrated by the fact already mentioned that this model is about 220 lb lighter than its predecessor.

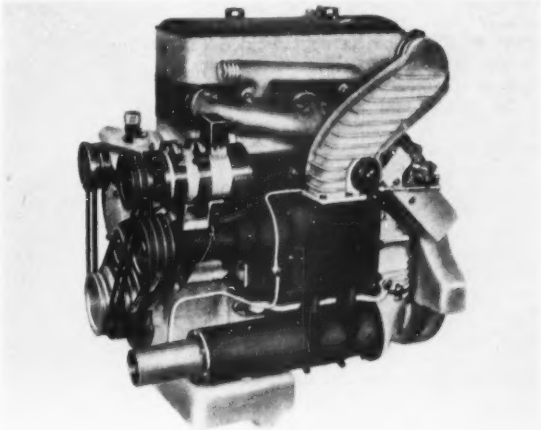
Both the side and the front cross members of the sub-frame are of inverted top hat section closed underneath to form a box section, by a plate spot welded to the flanges. The front cross member is about 4 in measured over the flanges, by 2 in deep; the side members are approximately 3½ in wide overall by 3 in deep at the front, tapering from the point where they pass under the toe board to a relatively small depth at the point where they terminate under the front seats.

Welded on the forward face of the front cross member, just inboard of its junction with the side member, are two brackets carrying the rubber mountings for the radiator. Also on the front face, but outboard of the junctions with the side members, are two more brackets to which the front end of the body shell is bolted. Two bolts passed vertically through each extremity of the cross member secure it to the bracing struts. These struts are of a complex cross section, of roughly top hat shape closed by a plate underneath. Each extends back over the cantilever suspension support, to which it is secured by three bolts, and then back to the dash. At this point it is supported by one more complex box section strut between the other side of the dash panel and the front pillar at its junction with the screen pillar.

A top hat section cross member is spot welded by its flanges to the front face of the dash panel between the bracing struts. A vertical bracing stay is fitted on each side between the top of the wheelarch and the drip channel round the bonnet opening. On the left, the stay also forms a back rest for the battery which is mounted in a tray overhung from and bracketed to the inner wall of the wheelarch.

Other attachment points between the

The 11·63 litre V8 engine as installed in the Saurer 6 H bus



The Saurer 5·82 litre four cylinder engine with the new blower

Darkened transparent panels are employed on each side of the ventilator duct on the roof of the Saurer 2H coach

sub-frame and the body structure are under the toeboard, where four bolts on each side are passed through a plate welded to the top of the side member. Two more bolts are passed through each of the flattened rear ends to attach them to a cross member under the forward edge of the front seats. Two more cross members are incorporated in the sub-frame, one spot welded underneath it between the cantilever suspension support and the rear lower wishbone bearing lugs. This member is a shallow inverted top hat section closed by a plate on top; its function is to act as a horizontal strut between the two suspension units. The other cross member is of inverted channel section to carry the rear engine mounting. At each end, this member is secured under the side members by two bolts.

The main side members of the body structure are formed by the sills. Their cross section is large, in comparison with normal practice where a separate frame is employed, because instead of a flat closing plate on the inner face, a top hat section on its side is employed. This top hat section is fabricated from two pieces. Its bottom flange, which together with the flat floor forms the lower wall of the section, is spot welded to the lower flange of the sill, while the upper and vertical walls of the section are formed by a separate plate suitably bent to shape. The upper flange of this plate is spot welded to the top flange of the sill, and its lower edge is flanged towards the centre of the vehicle and spot welded to the floor. The floor itself is extensively swaged to provide adequate stiffness.

At the front, the sill is attached to the dash side, which is stiffened by swages of large cross section pressed in the panel. The junctions of the dash side with the dash front panel and the front pillar form the edge members to take the end loads due to the transference of the bending from the sill to the forward structure.

Two cross members at the front carry the loads out to the sill. They both support the sub-frame, and the rear one carries the front seats. The foremost one is formed by welding under the front floor between the tunnel and the sill a small extension piece which is continued forward and then bent up vertically and flanged at the top to be spot welded to the toe board. This forms a closed triangular section member. The other cross member, under the forward edge of the front seat, is a simple top hat section reduced in depth and arched where it is continued over the tunnel.

At the rear, the attachment of the sill is not quite so simple, and more reliance is placed on the support afforded to the overall structure by the cant rails and roof. The extreme rear ends of the sill are flanged and spot welded to the front of the rear wheelarch and they are further supported by the rocker panel. Forward of this attachment is a cross member under the front edge of the rear seat. This cross member is arched and reduced in depth where it passes over the tunnel. It is of top hat section fabricated from two pieces joined together at the centre of the crown where each part is flanged up for spot welding. One of the upturned flanges is bent forward above the edge of the other and the seat pan is spot welded to it. The rim flanges of the top hat are attached to the floor and tunnel, and the member is secured at each end to the sill.

Another box section cross member is formed in the angle between the rising back of the seat pan and the horizontal boot floor. It is made simply by welding on a suitably flanged right angle section between the inner walls of the wheelarches. Since the seats are placed forward of the axle, the front ends of the wheelarches do not restrict the width at all but they do come slightly forward of the front face of the trim on the squab. This squab is a wire sprung frame resting back against a sheet of millboard supported at the top by an open box section cross member braced in three places by struts longitudinally positioned and attached at the rear to the drip channel at the top edge of the boot. The cross member is gusseted to the rear pillar about 1½ in above the waist.

The structure in the boot is simple, but well designed. A deep, rear cross member is formed below the lower edge of the boot opening by spot welding on a closing plate between the drip channel and the turned-in and flanged lower edge of the skirt panel. This cross member is extended round until it meets the rear of the wheelarches which, like those at the front, are spot welded at the wing cutting line to the outer wing or quarter panel. The boot floor is attached to the top of the drip channel, and a rubber section is fitted over the edge to seat the boot lid and to hold down the rubber covering on the boot floor. Under this covering is a sheet of felt impregnated with bitumen. The swages in the boot floor are filled with cork strip so that the covering lays flat.

Two longitudinal members complete the structure under the boot. They are of rectangular box section and are formed, one on each side in the angle between the floor and the inner wall of the wheelarch, by spot welding on a suitably flanged strip of angle section. At the rear they are gusseted to the rear cross member, and at the front they are welded to the cross member at the rear of the seat pan. The function of these members is to support the

An air compressor is mounted on the front of the Unimog dumper vehicle, and a side tipper trailer may be towed

spring shackle brackets which are attached beneath their rear ends. The front bracket for the spring eye is mounted beneath the cross member under the front of the rear seat pan.

Other new features

The modified Hillman Minx, Standard Vanguard and Triumph Roadster attracted a great deal of attention. However, they will not be described in detail here because the British cars will be fully described when they are shown at Earls Court.

Some modifications have been introduced in the Volkswagen, and a sectioned chassis was shown. On the engine installed in this chassis, a Solex downdraught carburettor incorporating a diaphragm accelerator pump is fitted. Behind the air cooled cylinders, ducts are provided to direct hot air into the body, and the flow is controlled by a lever operating a butterfly. On the suspension, the manufacturers state that the circular section torsion bars at the rear are made from a steel with a better fatigue resistance. Laminated torsion bars are still employed at the front. The telescopic shock absorbers are fitted between scissors links; in the latest version the length of the links has been increased so that the shock absorber is more nearly vertical and its effective leverage is greater. As a result, a lighter duty is imposed on the units.

When the 1953 Pontiac was announced a few months ago, much interest was aroused by the fact that a new "Curve Control" front suspension system was employed. This is a change in geometry rather than an entirely new system, and no new principles appear to have been developed. The relative lengths of the upper and lower links and the position of their pivot points have been arranged so that when cornering, the top of the outer front wheel moves inwards, instead of outwards, as in earlier models, relative to the bottom. In other words, the camber change is negative as the wheel rises from the normal $+\frac{1}{2}$ deg camber position. On rebound, or at the inner wheel when cornering, the camber change is positive. This means that the outer wheel, which takes the greater part of the cornering load, is in a more favourable attitude to react that load. How this arrangement affects other characteristics of the suspension is difficult to say without laying out the system in detail. It is not clear whether the positive and negative cambers, obtained respectively on the outer and inner sides, completely cancel out any tendency to a change in track. If they do not, it would appear that there might be a tendency towards a certain amount of tyre scrub, but in many quarters, this is not regarded as serious, since it is easily accommodated by the rolling motion of the wheel. Doubtless the designers are also satisfied that no undesirably large gyroscopic effects are introduced.

Another feature of this suspension system is that the whole unit on each side is tilted back 4 deg. In other words, a line joining the axes of either the two lower or upper wishbone bearings is set at an angle of 4 deg from the horizontal. The castor angle, however, is zero. The reason for the incorporation of this feature is that severe bump loads are not truly vertical since they have a rearward or drag component. The 4 deg angular setting is regarded as a good compromise for reacting the drag component through the spring instead of directly to the frame through the inner wishbone bearings.

The fire tender and snow plough adaptations of the Unimog

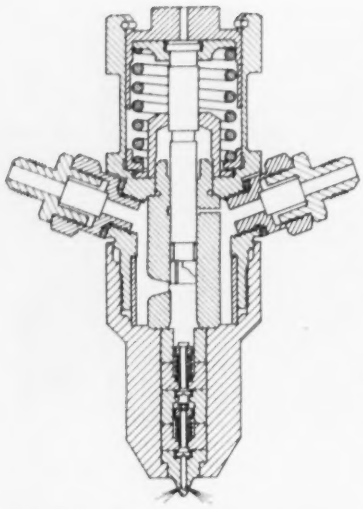
The Commercial Vehicle section

One of the most interesting stands in the Commercial Vehicle section was that of the Société Anonyme Adolphe Saurer. An unusual exhibit was the type 6H passenger vehicle. This has been designed specifically to replace an existing train service, or rather a tram which draws several coaches and trucks.

The vehicle comprises a Saurer rear engined bus together with two trailers made by Gabr. Moser et Cie., of Berne. One of the trailers is a passenger-carrying vehicle, and the other is for baggage. As an alternative, a 10 ton side tipper goods trailer can be substituted for the baggage one. The bus is about 39 ft 4 in long and its wheelbase is 19 ft $8\frac{1}{2}$ in, while the overall width is 8 ft $2\frac{1}{2}$ in. Its maximum permissible laden weight is 14·8 tons. Seats are provided for 40 or 46 passengers and in addition there is room for about 36 more to stand. The passenger trailer carries seats for 40, and there is room for another 60 standing. Its unladen weight is approximately 5 tons. The baggage trailer is attached by a draw-bar to the rear of the passenger trailer, and its weight is 1,800 lb unladen. The overall length of the train, with the lengths of the draw-bars taken into consideration, must be appreciably more than 100 ft. All wheels of the trailers are steered by a linkage connected to the draw-bars, so that the whole train runs on the same track.

The power unit is the direct injection V8 blown engine of 11·63 litres, and it is mounted longitudinally in the centre of the chassis at the rear. It has a bore and stroke of 115 mm by 140 mm and develops 200 b.h.p. at 2,000 r.p.m. The combustion chambers are of the double turbulence type.

The power unit driving the snow plough on the Unimog is mounted in the back, and the drive shaft passes through the cab



A sketch of the Chevrolet injection pump and spray nozzle

A twin disc dry plate clutch is employed, together with an 8-speed gearbox. Six of the speeds are synchromesh. This arrangement has been arrived at by employing a normal four-speed box with a two-speed auxiliary train incorporated, so that the two non-synchromesh speeds are the bottom gear with the two different auxiliary ratios. The auxiliary train is engaged by a pneumatic control, working in conjunction with a synchromesh mechanism. Its change speed lever is on the steering column, whereas the four-speed lever is mounted on the floor in the usual manner. A noteworthy feature of the system is that a pre-selector device is incorporated in the auxiliary change mechanism, so that with any one of the four speeds engaged, a second ratio is available instantly, simply by pressing the clutch pedal and releasing it to change the auxiliary ratio to the second speed, which has been preselected by means of the steering column control.

The hand brake controls the mechanical rear wheel brakes of the bus and the pneumatic brakes on the trailers. Operation of the pedal, on the other hand, applies the brakes pneumatically on all wheels of the bus and trailers. In addition, an exhaust brake is fitted.

Another interesting model shown by Saurer was their 2H rear-engined coach. This vehicle was outside the Show, where it was available for demonstration runs. It has a wheelbase of about 13 ft 6 in, and its overall length is 29 ft 8½ in, while its overall width is 7 ft 5 in. It can be supplied as a 21, 26 or 30 seater vehicle. The kerb weight is about 4¾ tons.

The chassis is the somewhat uncommon but well known Saurer design, which consists of a central tubular backbone of large cross sectional area with fork extensions forward and to the rear to carry the axles and other mechanical components. An advantage of this type of construction is that large luggage lockers can be incorporated in the unobstructed spaces on each side of the backbone member, beneath the floor. On the 2H, in addition to these lockers, there is space for more luggage on each side of the engine at the rear.

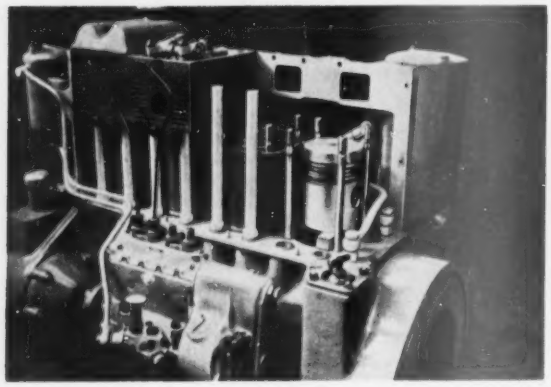
The power unit is the Saurer CR2D-Lm, four-cylinder diesel engine. It has a bore and stroke of 115 mm by 140 mm, and a swept volume of 5·82 litres. At 1,900 r.p.m., 110 b.h.p. is developed and the maximum torque is 340 lb-ft at 1,400 r.p.m. This engine, which is positioned vertically in the plane of the longitudinal axis of the vehicle, is a development of an earlier one, but the bore has been increased from 110 mm. As a result of this increase, it has been necessary to stiffen the head by incorporating internally some ribs and vertical stays. In addition to balancing the crankshaft, a vibration damper has also been fitted. Because of the increased load imposed on the crankcase by the new blower, further modifications have had to be incorporated in the crankcase to provide adequate strength and stiffness.

An integral cylinder block and crankcase of light alloy is employed, and wet liners are fitted. Double turbulence combustion chambers are incorporated. Light alloy pistons are used and the top compression ring is chromium plated. In the cylinder head, overhead valves are fitted and they are operated by the usual push rod and rocker mechanism. The injection pump and nozzles are of Saurer design and a mechanical fuel lift pump is used.

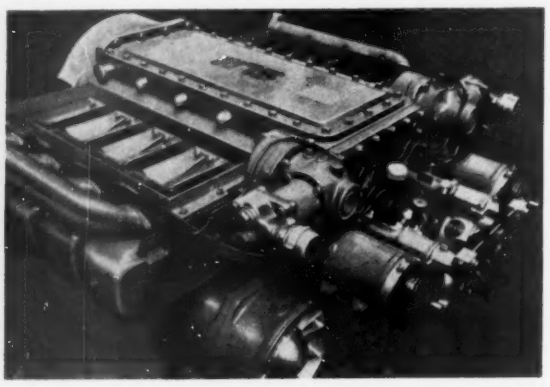
Few details are available on the new blower. It is of an improved design which can be run at an appreciably lower speed than the conventional Roots type unit. In this installation, at 1,900 engine r.p.m., the blower runs at 2,600 r.p.m. The two rotors are basically cylindrical in shape, one having three lobes equally spaced around its periphery, while the other has three grooves in which the lobes fit as the two rotors turn together. The lobes and the grooves are more or less semi-circular in cross section, but they are arranged helically along the length of the cylinder. The engine has a three point rubber mounting system. It is in a compartment between the two luggage lockers at the rear and is easily accessible through large doors in the rear quarters, which open into the lockers. Panels extending over most of the side wall area on each side of the engine compartment may be removed for access to the power unit. All the walls of the engine compartment and the removable panels are lagged with almost an inch thickness of asbestos cement covered with mill board.

A single dry plate clutch is employed. It is operated by a hydraulic servo mechanism. The gearbox is the Saurer 2C unit, which has four speeds and a two-speed auxiliary train. It is operated in the same way as that in the bus already described. A simple hypoid rear axle is fitted and the final drive ratio is 7 : 33.

Semi-elliptic springs with telescopic shock absorbers are employed front and rear. A worm and segment steering gear is installed. A truly remarkable feature of this coach is the small turning circle, which is only 22 ft. This is most essential in vehicles intended for touring in mountainous districts where sharp hairpin bends are frequently encountered.



The induction system and oil cooler installation on this vertical air-cooled engine fitted to the Mowag truck is similar to that on the flat engine



All the auxiliary components are duplicated and grouped at the front end of the SLM eight cylinder, horizontally opposed air-cooled engine for the Mowag truck

The coachwork is extremely well executed. The body panels are designed to help carry the loads and to provide a high proportion of the total stiffness of the vehicle. Despite this, the pillars are unusually small and all-round visibility is excellent.

Another feature that is most desirable in mountainous countries is the transparent panelling on each side of the centre third of the roof. Passengers greatly appreciate this feature as it enables them to see without difficulty the mountain tops all around them. It is also appreciated when travelling in cities, since tall buildings of interest on each side can be seen easily. In this vehicle darkened glass is used for the panels so as to cut down the glare of the sun and to screen passengers from intense heat. This, it is claimed, is better than using blinds for this purpose, because it eliminates flapping and rattling. Moreover, in Switzerland and many other countries, the brilliant sunshine can cause deterioration of blinds in as short a period as one season.

The centre third of the roof panel incorporates the intake and exhaust ducts of the ventilating system. A separate heating system is also installed. It consists of a diesel oil burner which is ignited by a glow plug. Diesel fuel is fed to the burner by an electric pump, and a fan drives the air past the heater and into ducts along each side of the vehicle just above floor level.

Operators who have tried this coach on long distance runs through difficult country have, it is claimed, been most agreeably surprised at its performance. Because of its very efficient brakes and good acceleration, coupled with its unusually small turning circle, extremely high average speeds have been easily maintained, and the amenities for passenger comfort leave little to be desired.

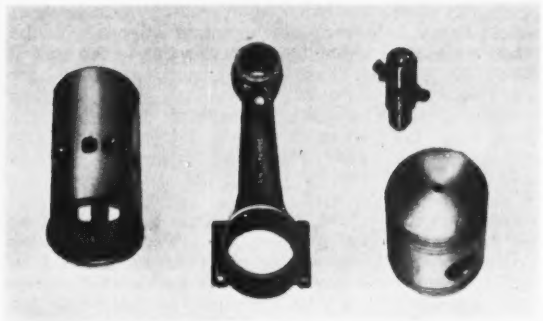
From experience of the vehicle's performance on the demonstration run, these claims appear to be well founded. Hairpin bends were taken at speeds which in any other vehicle would have been hair-raising if not disastrous. Even more spectacular was the way in which this coach was turned round and round several times in the road without reversing.

Despite the fact that a gale was blowing at the time of the demonstration and the wind was very cold, there was no draught inside the vehicle and the temperature was comfortably warm for passengers without overcoats. The seats also are comfortable. Each has a hand lever on the side to release it so that it can be slid away from the sides into the centre gangway of the vehicle to give all the passengers more elbow room; by means of the same lever the position of the squab can be adjusted. Another remarkable feature of the vehicle is the almost complete absence of engine noise, including diesel knock, although there was a certain amount of axle noise and exhaust silencer throb, both of which the manufacturers state will be cured by fitting modified components.

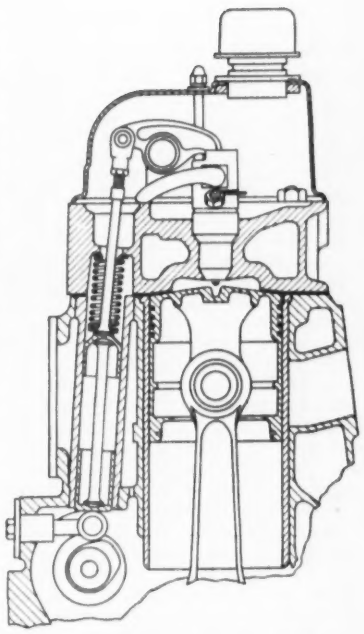
It was interesting to see how many variants there are of the Unimog Mercedes Benz, four-wheel drive, cross country vehicle. Contrary to general belief, this chassis is a post-war design. The manufacturers state that in the difficult period following the cessation of hostilities, it was decided that the wisest course to follow would be to take advantage of their experience in the design of military vehicles and produce a chassis suitable for cross country work, mainly for agricultural applications. There was a great deal of leeway to make up in connection with farm mechanization and it was thought that food production was bound to be one of the major post-war problems. Accordingly, it seemed that there would be a steady demand for this type of vehicle.

More recently, many other applications have been found for this chassis. Among those shown was a small fire tender, two variants of a snow plough vehicle, and a dumper with trailer for road work. The fire engine has a pump mounted above the bumper at the front, and ladders are carried on a frame at the back. Also on the rear portion are two bench type seats, positioned in the centre underneath the ladder and facing outwards, and at the extreme rear, the hoses. Other equipment, such as nozzles, etc., are strapped on or carried in lockers.

The snow plough vehicle can be fitted with two operating heads, one for soft freshly fallen snow, and the other for hard snow. Both may be readily attached or removed from the front end. The unit for soft snow is of the turbine type. Its essential parts are two rotating fan elements which face forwards and shoot the snow up a large diameter duct and away on each side of the vehicle. The other unit is much more robustly constructed and it is claimed that it will shift snow that is packed almost as hard as ice. It consists of a rotating drum fitted with heavy gauge vanes. This unit can be seen in the illustration. The direction of rotation of the drum is such that it drags the snow to the rear and then throws it upwards, whence it is discharged through funnel shaped ducts and thrown to each side.



Some components of the new Chevrolet two-stroke diesel



The rocker gear for the combined injection pump and spray nozzle in the Chevrolet two-stroke diesel

Undoubtedly a vehicle such as the Unimog that is designed for contractors would be most valuable in colonial territories where rough pioneer work on road making, etc., is carried out. A dumper body is fitted on the rear of the chassis, and mounted above the bumper at the front is an air compressor. On the vehicle displayed at the Show, one winch only was fitted, but the manufacturers intend to fit two in order to make the vehicle more suitable for lumber work. A side tipper trailer can also be supplied. Both the side tipper and the dumper are operated by pneumatic jacks.

The 4-4·5 ton Chevrolet truck is now offered with a new General Motors diesel engine. This unit is a four-cylinder, blown two-stroke engine, which in general resembles the earlier one, except that it is valveless. Its bore and stroke are 4·1 in by 4·1 in, and its swept volume is 216·5 cu in (3,546 cm³). The compression ratio is 18:1. The maximum b.h.p. is 90 and is developed at 2,800 r.p.m.; the maximum torque is 188 lb-ft at 1,800 r.p.m. The mean piston speed at 2,800 r.p.m. is 1,930 ft/min. With a dry weight of 975 lb, the weight: power ratio is just under 11 lb per b.h.p. An oil cooler is incorporated in the water jacket on

the side of the cylinder block, opposite the blower.

Although the overhead valve gear of the earlier engine has been eliminated, a camshaft, push rod and rocker mechanism still has to be incorporated to actuate the four combined injector pump and nozzle units mounted in the cylinder head. The mechanism is somewhat unusual in that a two-piece push rod is employed. The top of the upper portion of the push rod is pinned to the rocker end, and at the bottom a washer carries the return spring, the upper end of which bears against a shoulder in a hole in the lower face of the cylinder head. In this way, most of the rocker mechanism is carried on the head and the size of the spring in the injector is kept to a minimum.

The lower portion of the push rod seats at both its upper and lower ends in a piston type tappet. The bottom tappet rests on a roller follower interposed between it and the cam, and the upper tappet serves as a plain piston type guide and bears against the bottom end of the upper portion of the upper push rod. A horizontal link carries the roller, and is pinned to a block bolted to the side of the crankcase. This arrangement, as compared with a more conventional roller tappet, eliminates most of the side thrust on the tappet and reduces the load on the roller pin.

Another change that has been effected is in the combined injector pump and nozzle unit. This has been reduced in length from 6¼ in to approximately 4¼ in by employing a simple lever type control instead of the rack and pinion regulator which was formerly employed. Another interesting feature of the unit is that it is cooled by circulating a much larger quantity of fuel round the plunger barrel than is needed for delivery to the engine, and returning it to the tank. The nominal injection pressure is slightly more than 120 atmospheres.

A Roots type blower is employed, and the air passes to an induction chamber extending partly round the dry cylinder liner whence it passes into the cylinder through a number of small ports as they are uncovered by the piston. The exhaust gas is blown out through a smaller number of bigger ports on the opposite side of the liner. An interesting detail shown was the piston and small end bearings. Phosphor bronze bushes are pressed into the piston bosses and the small ends. These bushes have helical grooves on their bearing surfaces to spread the oil which is fed from the big end to the small end through a drilling up the connecting rod. Opinions are divided as to the desirability of grooving bearings in this way. Some claim that the grooves tend to break down the oil film.

A compact, air-cooled eight-cylinder horizontally opposed diesel engine for the Mowag truck was shown on the stand of Seitz and Ruf A.G. This unit was developed and is produced for the company by the Swiss Locomotive Works (S.L.M.). Its bore and stroke are 110 mm by 140 mm and the swept volume is 10·7 litres. The maximum power is 150 b.h.p. and it is developed at 2,200 r.p.m., and the torque is 427 lb-ft at 1,400 r.p.m. The overall dimensions of this engine are 51 in wide by 51 in long by 21 in deep.

The Warner electric brake

The direct injection principle is employed, and toroidal cavities are incorporated in the piston crowns. Each cylinder is a separate casting, with a detachable head. The fins on both the cylinder barrel and head are left as-cast. Overhead valves are incorporated. They are actuated by push rods and rockers served by two camshafts.

An unusual feature of this power unit is the extensive use of castings. Even the air baffles between the cylinders are cast. Large diameter cast induction trunks are mounted underneath the cylinders. At the front end is a blower which, although it supplies the induction system, is not a supercharger. Its main function is to force the air around the cylinders for cooling. This type of blower, which consists of a fan immediately behind a set of radially positioned guide vanes cast integrally with their sleeve housing, is suitable only for imparting velocity to the air and is not capable of generating appreciable pressure.

Air passes from the fan along the trunk, through an oil cooler, and then between the cylinders. Some of the air is by-passed without passing through the cooler, and is carried round a centrifuge chamber and thence into the integral induction manifold. A sectioned vertical engine which was also on the stand is illustrated together with the flat engine in order to show the arrangement of the induction system, which is much the same in both. It is stated that a great deal of development work has been done to determine the optimum blade and guide vane shapes and angles, and that the blower is now remarkably quiet in operation. Another noteworthy feature of this underfloor engine is that all the auxiliary components, such as the generator, injection pumps, etc., are duplicated so that the chance of a complete break-down is almost negligible. The two four-element injection pumps, together with the other auxiliaries are mounted on the front of the crankcase. In this position they are readily accessible, but the injection delivery pipes are rather long.

Among the accessories at the Salon was the Warner electric brake which is designed for trailers. Many advantages are claimed for this system. The trailer brakes are operated electrically and their action is synchronized with that of the tractor brakes. With electric operation there is no time lag, as there is with long pneumatic pipe lines on more conventional brakes, between the application of the tractor brakes and those on the trailer. The connections are more simple and less expensive than when pneumatic pipe lines are used. With this system it is possible to apply the trailer brakes independently of those on the tractor, a special hand-operated control unit being supplied for this operation. In the illustration, this unit is mounted low on the left-hand side of the stand.

Should the tractor and trailer accidentally become separated, the brakes of the trailer are applied automatically. This emergency operation is effected by means of a loosely suspended chain attached at one end to the tractor and at the other to a switch lever on the trailer. If the two vehicles part company, the chain pulls over the switch lever which brings a dry battery into circuit with the braking system and thereby stops the rear portion.

Another advantage of this electric operation is that the degree of braking applied at the trailer can be regulated according to the load. This is done by incorporating in the circuit a rheostat control which is mounted on the dash. The ease with which this adjustment can be made is a noteworthy feature. By regulating the brakes in proportion to the load a great deal of tyre wear, occasioned by the wheels locking when the trailer is lightly laden is avoided. At the same time fully effective braking may be obtained when heavily laden. In the illustration, the rheostat switch is mounted on a bracket to the right of the electric control box, above the brake.

The operation of the brake unit within the drum is simple. The linings are mounted on relatively flexible shoes which, when contact is made with the drum, have a servo action in the normal leading shoe manner. They are actuated by two cam levers mounted on the block between the ends of the shoes. A tension spring pulls the shoe ends against the levers, and prevents rattling when the brakes are not applied.

One or other of the levers, according to whether the vehicle is moving forwards or backwards, is actuated by a peg bearing against the end nearest the centre of the brake unit. This peg is carried on a circular electro-magnet, the magnetic forces of which act on a flat ring secured to the vertical face of

the drum. When the magnet is energized, it grips the drum, but not with a force sufficient in itself to have any noticeable braking effect. The contact, however, is firm enough partially to rotate the magnet, and the operating peg, which is attached to it, applies pressure to the end of the appropriate cam lever. The servo action of the brake shoe mechanism provides all the force necessary to maintain firm contact between the linings and the drum. A number of tension springs return the magnet ring and the brake shoes to their off position when the current is cut off.

Progressive braking action is obtained by the control unit, which in the mock-up shown on the stand was mounted above the brake. In this control a curved contact arm may be seen. When the control is applied, the arm swings upwards and contacts the ends of a number of copper strips mounted together on a block. The arrangement of these strips is such that the greater the movement of the control, the larger the number of strips contacted. In this way, a progressive action is obtained.

RHEOSTATIC BRAKES

An Improved Form for Trolley Buses

An improved form of rheostatic braking for trolley buses (British Patent No. 616679) has been developed by the General Electric Co. Ltd. It is claimed that the new form almost completely overcomes the disadvantages of previous schemes without losing any of their advantages. Early trolley buses were fitted with series-wound motors, and in some cases field-weakening was used to obtain top speeds. In few cases, if any, was an electric brake provided.

To overcome this, vehicles were fitted with compound-wound motors that permitted regeneration to line; they also had rheostatic braking. Large numbers of these equipments gave good service with materially reduced brake maintenance. It was found, however, that with the average driver regeneration was of very limited advantage for energy saving. In addition, even when skilfully used to effect a sensible saving of energy, regeneration often caused considerable trouble on a non-receptive line unless special absorption equipment was provided.

It must be remembered that the rheostatic brake has to operate through the transmission and must therefore be automatically limited to a safe maximum value. Series rheostatic braking as used on tramcars is technically unsuitable owing to its lack of limitation, but some equipments have been operated, either with multi-step speed relays of the governor type or forms of current control to prevent the torque exceeding a predetermined value.

Most commonly, compound-wound motors have been used. In most cases both fields were used for motoring, while the separately excited field was the prime excitation for the rheostatic brake. In some systems this field was used alone; in others it was associated with a few turns of the series field as inverse compounding; in still others it was associated with the braking resistors (starting resistors used as such) so that a form of inverse-compounding characteristic was obtained.

With all three schemes changes in the temperature of the motor field windings cause considerable variations in the braking effort. Since the limit of torque fixed for the transmission must not be exceeded in the cold condition, it follows that in the normal hot running condition the braking is not as powerful as it might be. These schemes also suffer from the fact that the braking effort is not maintained at lower speeds. Although this is no appreciable disadvantage so far as saving brake maintenance is concerned, it is not altogether satisfactory from the driver's viewpoint. Another disadvantage is that when the separately-excited field used for excitation is opposed by some turns of the series winding, the motor carries more copper than is required for the production of the magnetic flux.

The disadvantages outlined above are almost completely overcome by the new G.E.C. scheme. A series winding for motoring was considered to be best. If this were practicable, it was obvious that any separately-excited field winding would have to be kept to very small proportions to reduce to a minimum the amount of copper necessary. It was therefore arranged that a section of the series winding should be used for braking excitation, and that the limiting control should be by means of the separately-excited field winding. It then became obvious that if this field winding could function in both a positive and negative sense, not only could it be small, but few series turns would be needed for braking. A solution was reached by associating the controlling separately-excited field winding with the motor braking voltage.

The power circuits are so arranged that only a portion of the series field winding remains in circuit and the current flow relative to the armature is reversed. A braking circuit is therefore formed through the armature, the braking resistor and the series field winding. The speed-brake characteristic with this alone would not be suitable. However, it is so designed that at maximum build-up, it will not produce sufficient braking torque to damage the transmission although it may exceed a safe continuous running torque. This is necessary to avoid damage to the transmission in the unlikely event of failure of the regulating shunt field winding.

The regulating shunt field winding is energized in series with the motor armature across the line. Thus when the voltage of the armature is high, at high speed braking the armature voltage will exceed line voltage and create inverse excitation. At low speeds, the excitation will be cumulative owing to the fact that the line voltage will exceed the armature voltage. This gives a more constant tractive curve than the older systems, an advantage to the driver. By far the greatest advantage of he new scheme is that there is very little difference in torque between the cold and the hot state. Consequently, normal braking torques may be arranged equal to the safe running torque of the transmission. They are not in any way limited by considerations of the safety of the transmission in the cold state. (2041)

INDEXES AND BINDING CASES

Indexes and binding cases for volume XLII (January to December, 1952) of the "Automobile Engineer" are now ready. The price of the index is 7½d., and the binding case together with index is 6s. 5d., both post free. Alternatively, readers' copies can be bound in publishers' binding cases at an inclusive cost of 18s. 5d. including return postage. Remittance with order should be sent to the Publishing Department, Dorset House, Stamford Street, London, S.E.1.

Plastics Convention

AUTOMOBILE engineers are watching with interest the progress of reinforced plastics and their use as structural materials, particularly as components for automobile bodies. In the British Plastics Exhibition and Convention (June 8th-18th, Olympia, London), they will have the opportunity of hearing experts discuss properties and design considerations of these materials. The all-day session "Reinforced Plastics" will be held on June 10th at Olympia, and tickets (free on application) and details can be obtained from the Organizers, British Plastics Exhibition and Convention, Dorset House, Stamford St., London, S.E.1.

FUEL ECONOMY

The Influence of Ignition-Timing, Mixture Strength and Compression Ratio

By Dr. C. R. Webb and Dr. J. G. Lavender

THERE has been a growing tendency, within recent years, to assess the performance of an automobile petrol-engine in terms of fuel economy as much as in terms of maximum power output; and to-day the majority of road users, from fleet operators to private motorists, welcome any improvements in engine design, or changes in fuel quality, that decrease the fuel consumption of a vehicle.

Improvements in engine design may be divided into three groups: (a) design improvements that increase the maximum power output of a given capacity engine at the expense of fuel consumption; (b) improvements that increase the power output of an engine but do not adversely affect the economy; and (c) improvements that may slightly reduce the maximum power output but greatly increase the thermal efficiency and thereby reduce the specific fuel consumption.

This paper deals primarily with the last of these improvements and analyses the optimum adjustments of ignition, mixture strength and compression ratio, separately and conjunctly, that are necessary to obtain the lowest specific fuel consumption, or the highest brake thermal efficiency, over the entire range of speed and throttle openings. The changes in these adjustments required to produce maximum power, as compared with those for maximum economy, are also dealt with in some detail.

It is an unfortunate but inherent characteristic of carburettor-type petrol-engines that there exists only one combination of speed and throttle opening, within the entire working range, at which the engine will operate at maximum thermal efficiency or lowest specific consumption. This combination normally occurs about mid-way in the speed range and with slightly less than full-throttle opening.

Fig. 1 indicates part of a typical series of fuel consumption loops plotted against output torque over a complete range of speed and throttle openings, for a small, four-cylinder, side-valve automobile engine with a compression ratio of 6·5 to 1. The lines of constant specific consumption (lb/b.h.p.-hr) show the centre of the loops, or the maximum brake thermal efficiency, to occur at 2,700 r.p.m. with the throttle valve 55 per cent open. At speeds above 2,000 r.p.m. the maximum torque is given at full-throttle opening, whereas below 2,000 r.p.m. maximum torque is given with 55 per cent throttle opening. In general, it may be said that the engine operates most efficiently when developing between 80 per cent and 90 per cent of the maximum power output at any particular speed. As the throttle is closed to reduce the power output still further, the specific consumption rapidly increases and the engine becomes grossly inefficient. For example, at 2,700 r.p.m. with 55 per cent throttle opening the specific consumption is 0·535 lb/b.h.p.-hr or the brake thermal efficiency is 24·4 per cent, but with 15 per cent throttle opening at the same speed (not shown on Fig. 1) the specific consumption increases to 5·5 lb/b.h.p.-hr and the brake thermal efficiency is only 2·4 per cent.

On the level road this engine will drive a four-seater passenger car at 30-40 m.p.h. with a fuel consumption of 42 miles per gallon. Under such conditions, with the throttle partly closed to 20 per cent, the specific consumption is 0·80 lb/b.h.p.-hr, but if the specific consumption at this speed could be reduced to 0·535 lb/b.h.p. - hr, the vehicle would then have a fuel consumption of about 62 miles per gallon, an increase of 20 miles per gallon of petrol. An obvious solution would be to provide an engine of variable capacity such that the cylinders always work at "full-throttle" and consequently at the lower specific

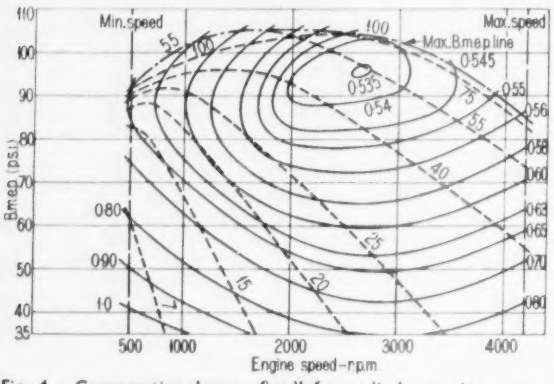


Fig. 1. Consumption loops. Small four-cylinder engine, compression ratio 6·5

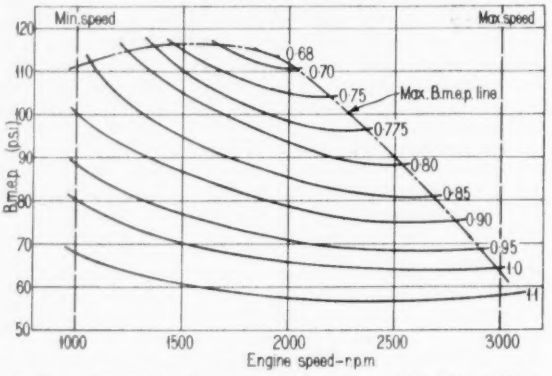


Fig. 2. Consumption loops. Fixed jet size and ignition setting. Compression ratio 5·5

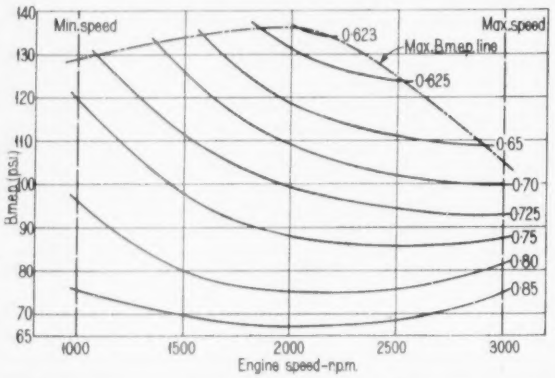


Fig. 3. Consumption loops. Fixed jet size and ignition setting, optimum compression ratio

TABLE I.
FIXED MIXTURE AND IGNITION SETTINGS, OPTIMUM COMPRESSION RATIO

Speed r.p.m.	Throttle Opening per cent	Compression Ratio	B.m.e.p. lb/in²	Air-Fuel Ratio	Spec. Cons. lb/b.h.p.-hr.
1,000	100	8·72	130·2	9·05	0·726
	60	8·65	128·1	9·16	0·726
	40	8·82	125·2	9·06	0·748
	30	9·11	121·0	9·04	0·761
	20	10·01	97·9	9·00	0·802
	10	14·30	61·3	9·00	0·818
3,000	100	10·85	105·5	11·74	0·667
	60	11·54	95·6	11·21	0·718
	40	12·82	71·4	10·65	0·941
	30	14·24	53·7	9·85	0·978

consumptions. Such an engine, although existing on the drawing board, is mechanically complicated and would require elaborate and expensive control gear. It may be mentioned at this stage that vehicles with excessively large capacity power units present an even worse picture than that quoted above, while heavy goods vehicles, with engines commonly working nearer to the maximum power output, give a somewhat better overall performance.

A further point arising from the loss of efficiency under part-throttle operation is that if any improvement that will increase (generally) the power output over the working range is made to an existing engine, and yet the engine is to be used to produce the same power output, then it follows that this output may be obtained with a smaller throttle opening. However, this reduction in throttle opening will generally cause a loss in efficiency, and it is quite possible that any reduction in fuel consumption will be outweighed by loss in efficiency and consequently, under similar conditions, the engine will operate less economically. At the same time, of course, the engine will have a far greater reserve of power.

The performance of a given fixed-capacity engine, operating under fixed conditions of speed and throttle opening, is dependent upon: (i) mixture strength, (ii) ignition advance, and (iii) compression ratio. Normally the mixture strength, or air-fuel ratio, is automatically controlled by the carburettor characteristics under different operating conditions, the only possible adjustment being the changing of the jet sizes. Standard carburettors, fitted to automobile engines to-day, usually have a characteristic that enriches the mixture as the speed decreases or the throttle is closed and weakens the mixture when the speed increases or the throttle is opened. It is an established fact that a mixture about 20 per cent rich (air-fuel ratio about 12 to 1) is required to produce maximum power conditions, and that a mixture 20 per cent weak is required for maximum economy. Thus a standard carburettor tends to produce maximum power output at the lower engine speeds and gives economy rather than power at the higher speeds. (This is borne out by the shape of the torque curve in Fig. 1). However, it must be realized that a mixture more than 20 per cent rich is extremely

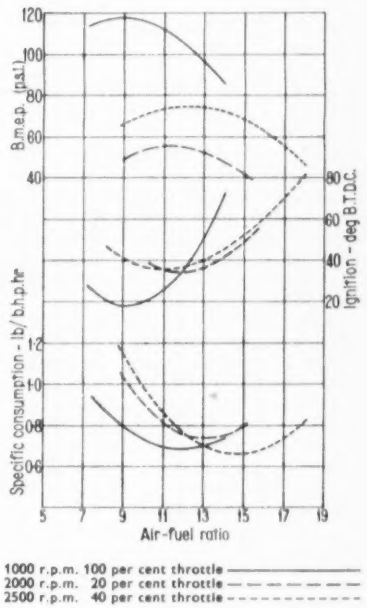

B.m.e.p. (p.s.i.)
Specific consumption - lb/b.h.p.hr
Ignition - deg B.T.D.C.
Air-fuel ratio
1000 r.p.m. 100 per cent throttle
2000 r.p.m. 20 per cent throttle
2500 r.p.m. 40 per cent throttle

Fig. 4. Variation of b.m.e.p., ignition setting and specific consumption with air-fuel ratio. Compression ratio 5·5

wasteful, for the power output decreases and the fuel consumption increases, thus giving a very low thermal efficiency. Similarly, for a mixture more than 20 per cent weak, the efficiency decreases owing to the considerable reduction in power output. Consequently, a carburettor should not under any circumstances produce a mixture strength outside the range of 20 per cent rich to 20 per cent weak. (Special considerations must be made for cold-starting and idling, for which purposes a choke or slow-running section of the carburettor is brought into operation to provide the extremely rich mixtures required.)

For each combination of engine speed and mixture strength there is a particular setting of the ignition timing that will give the maximum power output; whether maximum power or maximum economy is the criterion, the ignition should always be adjusted to this position. In general, the greater the engine speed or the weaker the mixture strength, the greater the ignition advance required. An attempt

TABLE II. COMPRESSION RATIO 5·5

Speed r.p.m.	Throttle Opening per cent	Air-Fuel Ratio	Ignition Advance B.T.D.C.	B.m.e.p. lb/in²	Spec. Cons. lb/b.h.p.-hr.
MAXIMUM POWER CONDITIONS					
3000	100	11·75	38	90·0	0·780
2500	100	13·2	31	109·0	0·640
2500	60	12·8	35	96·5	0·655
2500	40	12·3	37	74·0	0·740
2000	100	11·9	25	122·5	0·610
2000	60	12·2	24	120·0	0·595
2000	40	12·7	33	105·0	0·575
2000	20	11·3	34	55·0	0·800
1500	100	10·4	27	122·5	0·700
1500	60	10·6	23	121·5	0·695
1500	40	11·4	41	105·0	0·660
1500	20	11·1	40	69·0	0·765
1000	100	9·1	17	117·0	0·790
MAXIMUM ECONOMY CONDITIONS					
3000	100	15·1	63	79·5	0·660
2500	100	16·2	48	95·0	0·565
2500	60	15·9	54	89·0	0·575
2500	40	14·8	50	69·5	0·655
2000	100	15·5	49	111·0	0·530
2000	60	15·7	39	109·0	0·525
2000	40	14·0	42	103·0	0·530
2000	20	13·1	36	51·0	0·740
1500	100	13·1	41	110·0	0·580
1500	60	13·5	37	117·0	0·600
1500	40	13·4	51	99·5	0·595
1500	20	12·5	47	65·0	0·720
1000	100	11·8	32	107·0	0·680

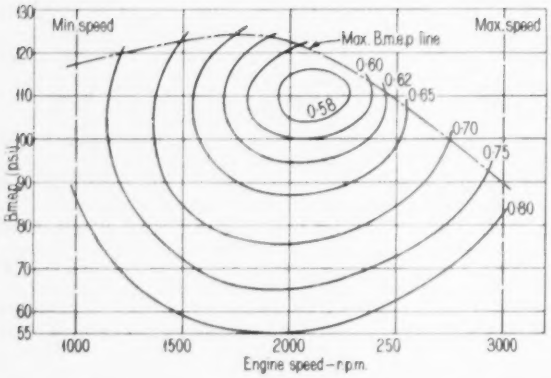


Fig. 5. Consumption loops. Maximum power, compression ratio 5·5

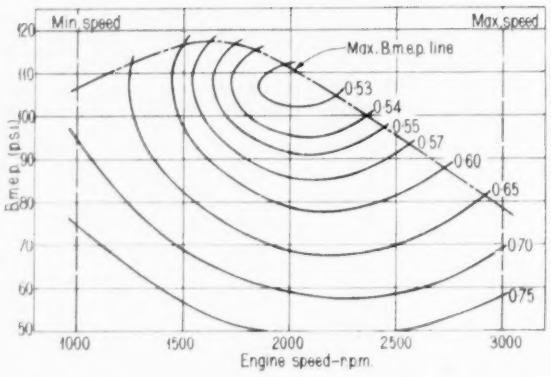


Fig. 6. Consumption loops. Maximum economy, compression ratio 5·5

is usually made to control the ignition advance automatically by means of centrifugal and vacuumatic devices, but of course usually without careful and individual design to suit both engine and carburettor. These arrangements cannot be expected to deal accurately with the combined variations of speed and mixture strength encountered over the entire working range.

The compression ratio is normally fixed by the manufacturers so that neither "pinking" nor detonation occurs under full-throttle conditions at the lowest operational engine speed, when running on a given grade of fuel. As the engine speed increases, however, or the throttle is closed, the octane requirement decreases, so that an increase in compression ratio could be accommodated without the incurrence of detonation. Consequential to increasing the compression ratio, the power output would increase; and thus to operate the engine at the greatest power output over the entire working range, the compression ratio would have to be changed continuously to suit the particular combinations of speed and throttle opening.

Test engine and measurements

With these considerations in mind, the tests described below were carried out on a Ricardo E.6/R variable-compression engine using 85-octane fuel (H.C.V. 19,500 B.Th.U/lb). Adjustments of ignition, mixture strength and compression ratio could be made with the engine running on-load:—

The test-engine details were as follows:—

Single cylinder; Poppet valves;
Bore 3·000 in; Stroke 4·375 in;
Compression ratio 4·5 to 20;
Valve timing:
Inlet, opens 8 deg B.T.D.C.,
closes 35 deg A.B.D.C.,
Exhaust, opens 46 deg B.B.D.C.,
closes 6 deg A.T.D.C.

The engine was fitted with a standard down - draught carburettor with a variable main jet, and was direct-coupled to a swinging-field dynamometer.

The readings taken were: —

(i) Speed r.p.m.
(ii) B.m.e.p. (torque) lb/in².
(iii) Fuel consumption, lb/hour.
(iv) Air consumption, lb/hour.
(v) Ignition timing, deg B.T.D.C.
(vi) Throttle opening percentage of maximum opening.
(vii) Compression ratio.

The fuel consumption was measured by means of a liquid flowmeter (Ref. 1) and an Alcock viscous-flow air meter was used to determine the air consumption. The ignition timing was measured stroboscopically, no reliance being accorded to the static calibration given on the magneto. The air-fuel ratio was calculated from the air and fuel consumptions readings. The b.h.p. was calculated from the b.m.e.p. and the engine speed, and the specific fuel consumption (lb/b.h.p./hr) was calculated from the actual fuel consumption (lb/hr) and the b.h.p. Figures of brake thermal efficiency, where quoted, have been calculated on the basis of the higher calorific value of the fuel, 19,500 B.Th.U/lb.

Description of tests

A series of tests was first made with a fixed compression ratio of 5·5 and a fixed ignition advance of 10 deg. A low compression ratio of 5·5 was selected so that in later tests, the ignition advance and mixture strength could be altered over a wide range without producing detonation. The variable main jet on the carburettor was adjusted to give a mixture strength producing maximum power at 2,000 r.p.m. and 100 per cent throttle opening. With these fixed settings, tests were made over the entire range of speeds and throttle openings. The results are shown graphically in Fig. 2, where b.m.e.p. is plotted against speed over the speed range 1,000 to 3,000 r.p.m. This figure is bounded by the maximum torque line, not necessarily obtained at 100 per cent throttle opening. Loops of constant specific con-

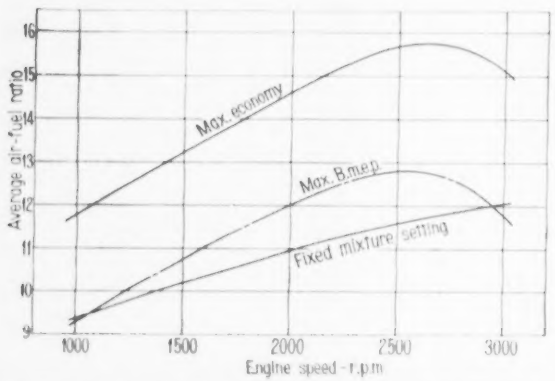


Fig. 7. Variation of air-fuel ratio with engine speed, compression ratio 5·5

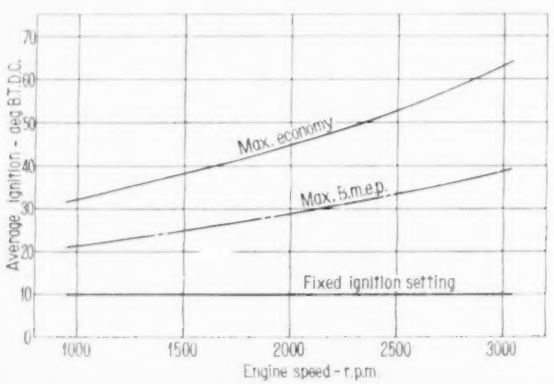


Fig. 8. Variation of ignition setting with engine speed. Compression ratio 5·5

sumption are drawn on the figure, and this quantity is seen to be a minimum at 0·68 lb/b.h.p.-hr at the maximum b.m.e.p. of 114 lb/in² at 1,900 r.p.m. The specific consumption rapidly increases from this point as the speed or throttle opening is varied.

With the same settings of the carburettor and ignition, a test was next carried out in which the compression ratio was adjusted to the maximum permissible value at each combination of speed and throttle opening. At the lower end of the speed range and at the wider throttle openings, the possible increase in compression ratio was limited by the onset of detonation, but at the higher speeds and smaller throttle openings detonation did not occur even with compression ratios as great as 14 to 1. In such cases the setting of the compression ratio was

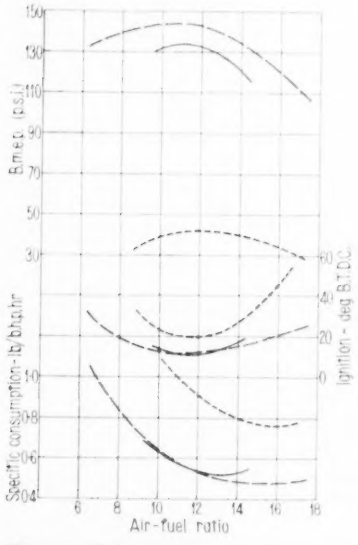

B.m.e.p. (p.s.i.)
Specific consumption - lb/b.h.p.hr
Ignition - deg B.T.D.C.
Air-fuel ratio
1000 r.p.m. 40 per cent throttle
1500 r.p.m. 100 per cent throttle
2500 r.p.m. 20 per cent throttle

Fig. 9. Variation of b.m.e.p., ignition setting and specific consumption with air-fuel ratio. Compression ratio 7·8

adjusted to produce maximum power output. (It was noticed that above compression ratios of about 12 to 1 the net power output tended to decrease as a result of the excessive negative work.)

The results of the test are shown graphically in Fig. 3. Table I indicates the variation of compression ratio used at 1,000 r.p.m. and 3,000 r.p.m. The consumption loops will be seen to be of a similar shape to those in Fig. 2, but by resetting the compression ratio at each combination of speed and throttle opening, the maximum torque line has been raised by about 18 per cent. Also, although the minimum specific consumption has been reduced only from 0·680 to 0·623, the specific consumption under part-throttle conditions has been improved by a rather greater amount. For instance, with a fixed compression ratio of 5·5 at 2,000 r.p.m., a b.m.e.p. of 90 lb/in² was

TABLE III. COMPRESSION RATIO 7·8
MAXIMUM POWER CONDITIONS

Speed r.p.m.	Throttle Opening per cent	Air-Fuel Ratio	Ignition Advance B.T.D.C.	B.m.e.p. lb/in²	Spec. Cons. lb/b.h.p.-hr.
3000	100	12·2	21	111·1	0·605
	60	12·4	26	106·3	0·610
	40	12·4	39	75·7	0·675
2500	100	11·7	21	123·0	0·595
	60	11·5	25	112·5	0·620
	40	12·2	21	85·4	0·660
	20	12·2	20	42·1	0·890
2000	100	11·8	18	142·1	0·530
	60	12·7	14	137·0	0·510
	40	12·0	23	119·3	0·550
	20	12·0	25	68·3	0·645
1500	100	11·0	12	144·1	0·575
	60	12·5	11	142·6	0·510
	40	11·5	12	133·4	0·565
	20	12·0	16	84·1	0·600
1000	100	11·25	12	139·0	0·560
	60	12·0	10	135·0	0·542
	40	11·25	10	134·1	0·562
	20	12·2	14	106·3	0·555

obtained with a specific consumption of 0·82, whereas the same b.m.e.p. at 2,000 r.p.m. was obtained with a specific consumption of 0·74 when working with the optimum compression ratio.

A further series of tests was carried out with a fixed compression ratio of 5·5, but at each speed and throttle opening separate tests were made over a wide range of air-fuel ratios. In every test the ignition was set to produce the maximum b.m.e.p. Fig. 4 shows some sample curves drawn for particular speed-throttle combinations, where b.m.e.p., ignition advance, and specific consumption are plotted against a range of air-fuel ratios. In all cases each quantity passes through a maximum or a minimum point at a particular air-fuel ratio. The maximum values on the b.m.e.p. curves indicate the air-fuel ratio required for maximum power, the ignition advance needed and the resulting specific consumption being read directly from the other curves. Similarly the minimum on the specific consumption curve indicates the air-fuel ratio required for maximum economy, and the corresponding values of b.m.e.p. and ignition can be read. The curves of Fig. 4 are typical of all those drawn, the general tendency being as follows: At a very rich mixture, the b.m.e.p. is low, specific consumption high, and the ignition fairly well advanced. At an air-fuel ratio in the region of 12 (but this figure can vary between 10 and 13), the maximum b.m.e.p. is obtained with the ignition at very nearly the minimum angle of advance. As the mixture is further weakened the b.m.e.p. decreases but the specific consumption decreases also to a minimum value (maximum economy) at an air-fuel ratio between 14 and 16, and the angle of ignition advance increases rapidly. Further weakening of the mixture produces a further decrease in b.m.e.p. but the specific consumption begins to rise again.

TABLE IV. COMPRESSION RATIO 7·8
MAXIMUM ECONOMY CONDITIONS

Speed r.p.m.	Throttle Opening per cent	Air-Fuel Ratio	Ignition Advance B.T.D.C.	B.m.e.p. lb/in²	Spec. Cons. lb/b.h.p.-hr.
3000	100	16·5	41	93·1	0·515
	60	16·4	43	87·2	0·520
	40	15·6	51	65·5	0·570
2500	100	16·0	29	109·0	0·480
	60	16·7	39	90·2	0·490
	40	15·7	38	77·0	0·515
	20	15·8	44	35·1	0·760
2000	100	16·0	27	127·2	0·425
	60	16·5	27	119·5	0·440
	40	16·3	37	102·0	0·470
	20	14·8	37	58·0	0·575
1500	100	15·0	18	128·0	0·475
	60	15·0	14	125·5	0·465
	40	17·0	33	109·1	0·465
	20	15·2	30	73·5	0·515
1000	100	14·5	20	125·0	0·460
	60	14·5	16	125·3	0·460
	40	13·2	14	127·0	0·515
	20	15·2	26	91·5	0·520

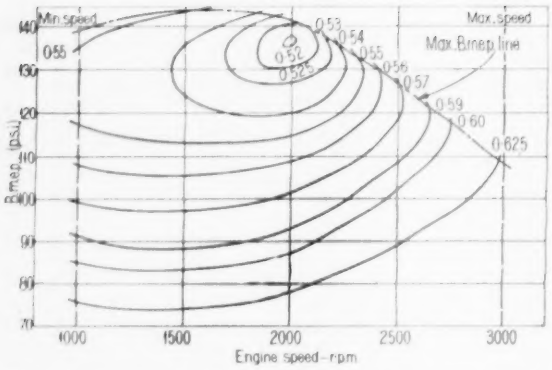


Fig. 10. Consumption loops. Maximum power, compression ratio 7·8

Table II has been compiled from a large number of curves similar to those of Fig. 4 and values of b.m.e.p., ignition advance, air-fuel ratio, and specific consumption are given for maximum power and maximum economy conditions over the entire range of speeds and throttle openings. Figs. 5 and 6 show graphically the consumption loops obtained when the engine is operated under maximum power and maximum economy conditions, respectively. The maximum b.m.e.p. line of Fig. 5 (maximum power) is, of course, higher than that of Fig. 6 (maximum economy) but both lines are higher than the maximum torque line of Fig. 2. Again, whereas the consumption loops of Fig. 6 are lower in value than those of Fig. 5, both sets of loops show a considerable improvement economically on Fig. 2. This test shows quite clearly the potentialities of being able to control the ignition setting and mixture strength to the optimum values throughout the working range of the engine.

Table II indicates that neither air-fuel ratio nor ignition advance is required to vary according to any recognizable laws, but some general conclusions may be drawn if both quantities are averaged for each speed. Figs. 7 and 8 show these average quantities plotted against speed for both maximum power and maximum economy conditions. Fig. 7 shows the variation in mixture strength required from the carburettor, and Fig. 8 the requirements of an automatic advance mechanism. Also plotted on Figs. 7 and 8 are the average values of the mixture at each speed as supplied by the carburettor in the first test and the constant value of ignition setting used. It is evident that better economy would have been obtained in the first test with an ignition advance of about 30 deg and a somewhat weaker mixture. The curves also indicate that both ignition control devices, mentioned earlier, could be designed and used in conjunction with each other to provide automatically the correct ignition settings between the extremes of the maximum economy and maximum power curves of Fig. 8.

An exploratory test was then made at 2,000 r.p.m. and 100 per cent throttle opening to find, by successive trial and error, the best combination of compression ratio, mixture strength and ignition advance to produce the absolute minimum specific consumption. This compression ratio was found to be 7·8 to 1. At higher compression ratios, although a greater power output was obtainable, it was found that the specific consumption was higher. It should be noted that at these high compression ratios, both enrichment of the mixture and ignition advance were limited by the onset of detonation, and consequently any increase in compression ratio had to be compensated by retarding the ignition or weakening the mixture strength.

Using the compression ratio of 7·8 a similar series of tests was made, as recorded above for a 5·5 compression ratio. Typical curves of b.m.e.p., ignition advance and specific consumption are plotted in Fig. 9 for particular speed-throttle combinations. Readings from all these curves of the b.m.e.p., ignition advance, mixture strength, and specific consumption have been tabulated in Tables III and IV for both maximum power and maximum economy conditions. Each group of figures was used to draw diagrams of b.m.e.p. against speed, showing the consumption loops. Fig. 10 is the diagram for maximum power, and Fig. 11 that for maximum economy.

Fig. 11 shows the minimum specific consumption to be 0·425 lb/b.h.p.-hr, corresponding to a brake thermal efficiency of 30·7 per cent. This value may be taken to represent the highest possible thermal efficiency for the particular fuel and design of engine and combustion chamber under test. It is of interest to note that even when operating the engine under maximum economy conditions with a compression ratio of 7·8 (Fig. 11), the maximum torque line is higher than for maximum power conditions at a compression ratio of 5·5 (Fig. 5). For maximum power conditions, compression ratio 7·8 (Fig. 10), the maximum torque line is considerably higher than for the optimum compression ratio with fixed mixture and ignition settings (Fig. 3).

Table III indicates that, contrary to the previous case, the air-fuel ratio, over the entire range of speed and throttle openings, requires to be nearly constant at a value of about 12 to 1 for maximum power and of 16 to 1 for maximum economy. The ignition advance, however, still shows some random variations, and in Fig. 12 average ignition settings are plotted for each speed corresponding to Fig. 8 for the lower compression ratio. It can be seen that 11 deg less advance at 1,000 r.p.m. and 14 deg at 3,000 r.p.m. are required, but the same kind of variation with engine speed and manifold depression is required as was the case with the lower compression ratio.

Conclusions

It follows that raising the compression ratio to 7·8 and resetting the mix-

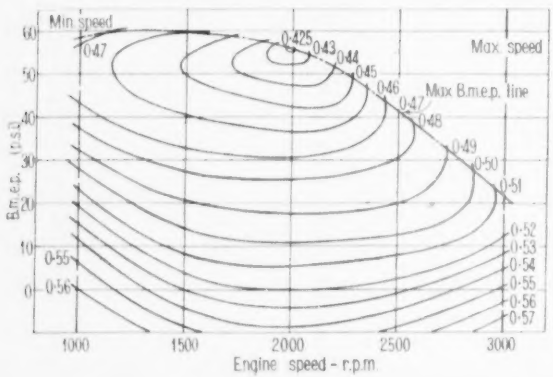


Fig. 11. Consumption loops. Maximum economy, compression ratio 7·8

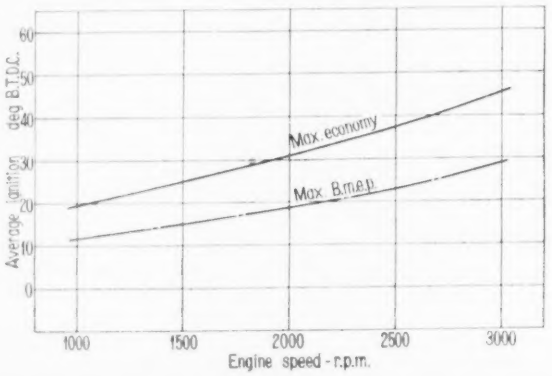


Fig. 12. Variation of ignition setting with engine speed, compression ratio 7·8

ture and ignition controls, is desirable from both the power output and the economy points of view. Where conditions are set for maximum economy, there still exists a 5 per cent gain in power output over the maximum power conditions at the lower compression ratio. At half full power at 2,000 r.p.m. the specific consumption is 0·54 (Fig. 11) corresponding to a brake thermal efficiency of 24·1 per cent, compared with a specific consumption of 1·05 or 12·4 per cent efficiency when giving the same power output at 2,000 r.p.m. in the initial test (Fig. 2). At smaller throttle openings the position becomes even more favourable.

A comparison of Figs. 1 and 2 shows that the consumption loops obtainable with the single-cylinder Ricardo engine are very similar in shape to those given by a standard multi-cylinder automobile engine, and consequently the conclusions arrived at in this investigation may be applied, quite accurately, to high-speed multi-cylinder engines. The inherent falling-off in brake thermal efficiency at lower throttle openings may be checked quite substantially. First, the best overall compression ratio to suit the fuel must be chosen, and it is seen that there is little to gain by fitting a mechanism to vary the compression ratio for different operating conditions. Then the ignition must be carefully set, and the automatic control devices must be designed to suit the particular engine and carburettor in question, and must follow curves similar to those in Fig. 12. One standard automatic control will not suit all engines and fuels. Finally, carburettor design must be improved to provide automatically a reliably constant mixture strength over the entire working range. The correct combination of settings is, of course, essential; for example, increasing the compression ratio with a rich mixture or the spark over-advanced would produce pinking and detonation, which would be deleterious as well as uneconomical. The ignition settings suggested should not impair cold starting, where, as previously stated, the carburettor must provide an extremely rich mixture.

None of the requirements mentioned is difficult to meet in practice, and in the authors' opinion, a considerable improvement in the fuel economy of present-day engines is still possible. The work described above was carried out in the laboratories of Queen Mary College (University of London).

Reference

1. J. G. LAVENDER and C. R. WEBB; "A Direct-Reading Low-Range Liquid Flowmeter," *Engineering*, Vol. 171, May 25, 1951, p. 619.

FLOW IN DIESEL COMBUSTION CHAMBERS

FLOW phenomena are discussed in an article in German entitled "Flow Phenomena in the Combustion Chambers of Diesel Engines," by H. Treibnigg, published in the October 1952 issue of *M.T.Z.* The paper deals with a mathematical investigation into flow phenomena in the combustion chamber of a direct injection diesel engine, and gives an account of tests with models.

The case of a piston head having a central cylindrical recess is considered. In the part of the cylinder above this recess, the degree of compression is smaller than that in the outer ring above the crown, and during the compression and expansion strokes, continuous balancing of pressure occurs between these two regions, with resultant flow processes taking place. Employing certain simplifying assumptions, the mathematical procedure is said to offer a rapid means of determining the main factors affecting flow speed in the cylindrical transitional cross section which varies by reason of the piston movement. Equations are derived, and their numerical evaluation is explained by means of an example. Graphs indicate the effects of varying several factors, so as to cover all cases coming within the practical range.

Flow tests were carried out on a model cylinder of 200 mm diameter arranged horizontally and sectioned so that it was open at the top. A crank mechanism actuated the piston, and the cylinder and piston heads, as well as stroke : bore compression ratios, could be changed readily. Water is used as the fluid, and the three dimensional flow conditions in actual cylinders were reduced to two dimensional flow. Powdered aluminium was used to make the flow processes visible, while the use of water instead of air permitted very low piston speeds and hence visual observation and photographic reproduction of the flow phenomena. A number of piston head shapes were investigated by the taking of cinematograph films and single photographs. Detailed explanations of these illustrations are given.

Results indicate that, where the cylinder and piston head shapes produce strong flow and large displacements of the fluid, these flow phenomena are local. In order to exploit this fully, fuel should be injected in these localities and the speed of the incoming fuel particles should be low enough for them to be taken up by the flow instead of breaking through it. Advantages are also to be gained from the use of combustion chambers of simple shape. (*M.I.R.A. Abstract No.* 6142.)

COACH HEATING EXPERIMENTS

TO provide information on the heating of P.S.V.s in sub-zero climates, an investigation was undertaken by Smith's Motor Accessories Ltd., using a "mobile laboratory" consisting of a 37-seater Duple coach body mounted on a Leyland Royal Tiger chassis and fitted with various types of heater with controls which allowed tests of independent systems to be made. Some utilized hot water from the engine system, while others were of the "combustion" type deriving heat from fuel oil. Both laboratory tests on the German wind-jet stand of Suddeutsche Kulerfabrik at Stuttgart, and road tests in the Austrian Alps were made, the latter being in atmospheres down to 5 deg F, and at altitudes up to 6,000 ft.

A description of the equipment and tentative results of the tests are given in an article in *Motor Body*, August, 1952. The tests indicated that heating by fresh air, either entirely or in part, is essential if misting and condensation on the interior face of the windows and windscreen is to be avoided. With a full load of passengers and with 20 deg of frost, a minimum of 400 ft^3/min of fresh air is desirable. The front face or external wall of the coach, immediately below the windscreen, was found particularly suitable for the air intake. For even distribution of air temperature, the use of two, or preferably four, heaters distributed fore and aft is desirable if elaborate ducting systems and powerful air-circulating equipment are to be avoided.

The heat loss from the coach, 30 ft long by 8 ft wide, was found to be approximately 18 B.Th.U/min/deg F temperature different between inside and outside. To cope with 20 deg of frost, the temperature differential should be 50 deg F, while for 32 deg it should be 60 deg F. This would require a heater input of 78,000 or 94,000 B.Th.U/hr respectively for an empty coach, though rather less when full of passengers. It is shown that the heat available from the engine cooling water is quite inadequate during low-throttle or down-hill driving. In these cases boosting by a "combustion" type heater was found necessary, and if this is used to heat the cooling water there is available the added advantage of a means of rapid warming up. (*M.I.R.A. Abstract No.* 6052.)

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Servicing Guide to British Motor Vehicles Volume II

By J. N. McHattie, A.M.I.Mech.E., M.S.A.E.

London: THE TRADER PUBLISHING CO. LTD., Dorset House, Stamford Street, London, S.E.1. 11 × 8¼. 193 pp. Price 42s.

This volume of the "Servicing Guide to British Motor Vehicles" contains information on 1951 and 1952 models. To meet the demand for specialized servicing information as quickly as possible on new models, this volume has been produced on a smaller scale than Volume I. It is divided into three sections, for cars, commercial vehicles and tractors. In each section the various makes are arranged alphabetically. Such popular and interesting models as the Austin "7" and Somerset, the Ford Consul and Zephyr, the Jaguar XK 120 and Mark VII, the "big" Bedford and the new Fordson Major tractor are included.

The data sheets which comprise this volume are arranged in exactly the same way as those in the first volume. They follow the same standardized layout, with sectional arrangement drawings of the principal components, sketches and tabulated data in a uniform style. Details of servicing operations, with stress laid on unusual features of design, are given in telegraphic style in the text. Stout thumb-indexed divider cards carry alphabetical lists of the contents of each section.

This publication is, of course, primarily intended for the information of trade repairers and service men. It should also appeal to students of design, since it affords a ready means for comparing a wide variety of designs. Since it is intended for use as a reference book, the volume has been produced to withstand wear and tear. It is bound in stout board covers finished in a special cloth that is impervious to grease and water. The strongest available paper has been used.

The information is presented in the most effective possible way. An introduction explains the system on which the information is presented, an illustrated glossary explains the terminology used throughout, and there is a list of alternative part names. An appendix gives a number of useful conversion tables for British, Continental and American units, along with other tabulated information of a general character.

Superchargers for Internal Combustion Engines (Ladeeinrichtungen fur Verbrennungs Motoren)

By Dr.-Ing. Bruno Eckert.

Stuttgart: FRANKH'SCHE VERLAGSHANDLUNG. 8¼ × 11¼. 63 pp. Price DM 13.2 (1 DM = 1s. 8d.).

The desire for increased specific output makes the use of superchargers increasingly popular, yet although a considerable number of papers were published on the performance of various blower types and the performance of supercharged engines, no concise text dealing with the performance of blowers, their size determination, matching of blowers and engine and their effect on engine performance was hitherto available. The present book written by a member of the Research Institute for Automobiles and Automobile Engines (FKFS) at the Technical High School in Stuttgart fills the gap in a most effective manner. After dealing briefly with the advantages and limits of supercharging, the author turns to basic considerations of compression with particular reference to the important pressure-density-temperature ratios relations. Following this, the air throughput of four and two stroke engines is considered.

Having dealt with the main factors determining the blower performance requirements, the author deals in detail with non-dimensional factors of importance for comparison and selection of suitable blowers. Here the chart of blower performance data plotted on the basis of discharge, pressure and throttling coefficients is particularly useful. The next chapter deals with the performance and size determinations of positive displacement blowers of the reciprocating type, as well as rotary units including Zoller, Lysholm and Roots designs. In every case, the performance considerations are followed by an example of size determinations.

Some twenty-two pages are devoted to the performance and design of centrifugal and axial blowers. Starting with the equation of continuity and Euler's equation, the author proceeds to deal in considerable detail with the effect of blade numbers and shape as well as diffusor design on blower performance, this being followed by a detailed example of rotor size and blade shape determination. The axial flow blower is dealt with in a similarly thorough manner embracing the basic theoretical considerations and concluding with a detailed example of size and blade data determinations. Here the performance of cascade of blades is based on earlier theoretical considerations due to Weinig, but the use of actual cascade performance data as for example determined by rig tests and published by N.G.T.E. should give more reliable results, particularly since the use of Weinig angle correction factors tends to give unduly high blade angle values at the root. This should not, however, detract from the value of the basic considerations, which in conjunction with the data published by Keller and Koffman should be also of value to fan designers.

The matching of blower delivery characteristics with engine air requirement is considered next, again, on the basis of worked out examples. The last fourteen pages are devoted to the exhaust driven turbo-charger. Here the matching of blower power demands with turbine output is considered in a manner which satisfies both theoretical and practical requirements. This is followed by chapters dealing with performance and size determination of impulse and reaction turbines, and here again, examples of size determinations are given. The concluding chapter deals with the basic performance of the Buchi system of turbo charging. The book includes eighty illustrations and six tables.

The clear and concise manner in which the generally complex matter is treated, augmented by numerous examples of size determinations makes this book eminently suitable to both students and practising engineers who will appreciate its immediate utility. Because of this, it will be a most valuable addition to the library of firms or individuals who have to deal with engine supercharging.

Metal Industry Handbook and Directory 1952

London: LOUIS CASSIER COMPANY LTD., for *Metal Industry*, Dorset House, Stamford Street, London, S.E.1. 9 × 6. 448 pp.

This Handbook and Directory, now in its 41st year of publication, is an indispensable reference work for those engaged in the production or the use of all non-ferrous metals. It has long been recognized as an invaluable source of information and reliable data. The work is divided into four sections: general properties of metals and alloys; general data and tables; electro-plating; and the directory.

Official British specifications for non-ferrous alloys have been brought up-to-date, and the section devoted to proprietary alloys has been completely revised. The directory for buyers has been greatly enlarged in order to facilitate the task of finding suppliers of plant for, and products of, the non-ferrous metals industry. This publication is available only with the weekly journal *Metal Industry* at a combined subscription of £3 5s. 0d. per annum.

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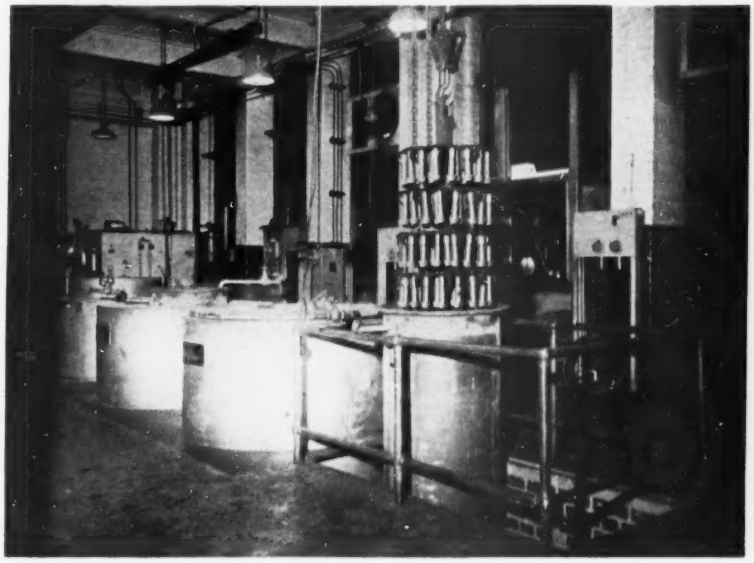
GAS CARBURIZING

Methods Employed by The Rover Co. Ltd. for Crown Wheels and Pinions

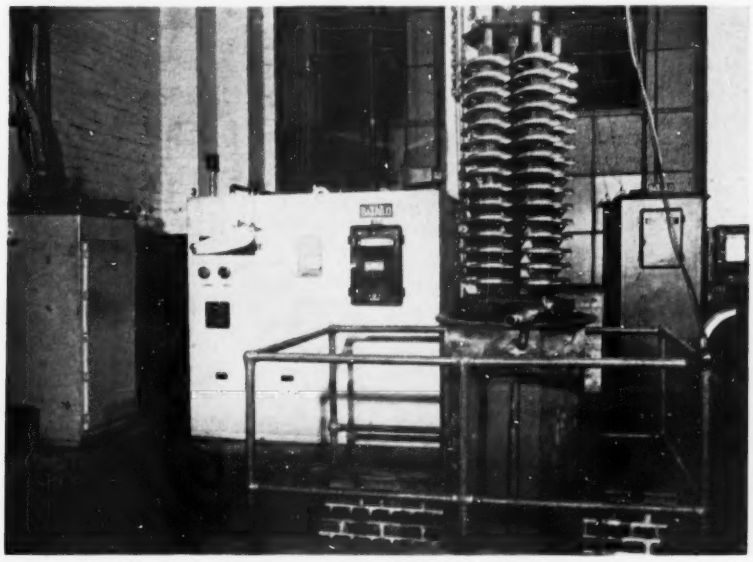
DESPITE the many advantages to be gained from the use of gas carburizing in appropriate applications, there is still in many quarters a hesitancy to adopt the process. Admittedly, considerable capital outlay is entailed for the installation of gas carburizing equipment, but the benefits to be gained are such that there is no doubt that this expenditure will often lead to considerable production economies for equal, and more probably, an improved standard of work. This has been fully proved by The Rover Co. Ltd.

Of all the transmission components in a motor vehicle, the crown wheel and pinion are subject to the most arduous duties. Every aspect of their production must be closely controlled if they are to give satisfactory service. Scrupulous mechanical accuracy in the production of these components is a fundamental of satisfactory performance, and the heat-treatment carried out on these parts is of equal importance. It must produce a case of the correct hardness, depth and composition, and at the same time distortion must be reduced to the minimum possible amount. The experience of The Rover Co. Ltd. in treating crown wheels and pinions is that gas carburizing gives more constant results in these respects and is also more economical than the pack hardening that was previously employed.



Wild-Barfield gas carburising plant at the works of The Rover Co. Ltd. A charge of bevel pinions is being withdrawn from the retort

Before the equipment and the technique employed are discussed, brief mention may be made of procedures prior to the carburizing and hardening operations. To begin with, the forgings from which the crown wheels and pinions are produced are themselves produced from steel billets that have passed a rigorous inspection for chemical analysis, mechanical properties and homogeneity. They are made with generous machining allowances, and before any machining is carried out they are given a cyclic annealing that not only relieves all forging stresses but also produces the optimum structure for gear cutting.

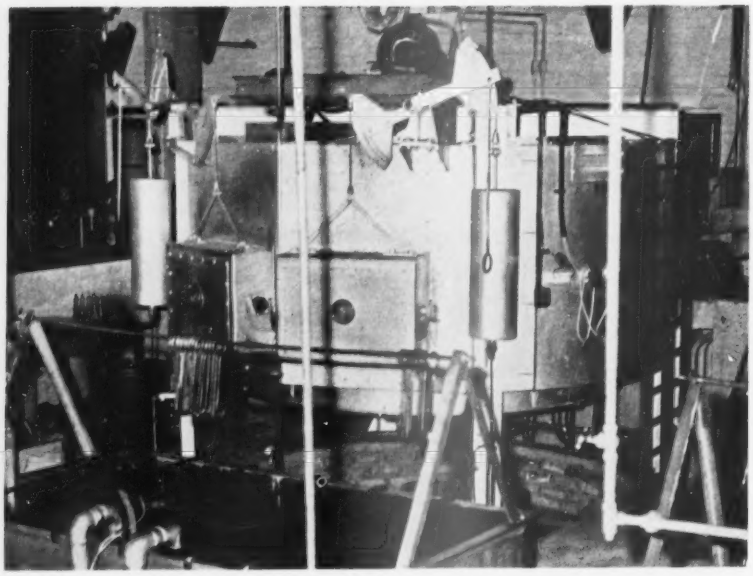


Loading a retort with crown wheels. The gas preparation plant is shown in the left background

Gas carburizing equipment

When the Company decided to adopt the gas carburizing process, an exhaustive study was made of the various equipments that were available, and finally the prepared town's gas carburizing equipment manufactured by Wild-Barfield Electric Furnaces Ltd., Elecfurn Works, Watford By-Pass, Watford, Herts, was chosen as the most suitable for the desired purpose.

Town gas as supplied from the mains is not suitable as a carburizing medium. The prime carburizing constituents of raw town gas, methane and carbon monoxide, are ideal since they do not leave a hard deposit on the surface of the steel at temperature, but there are other constituents that make town gas in its raw state unsuitable as a carburizing medium. They are carbon dioxide, oxygen, water vapour and small amounts of organic sulphur compounds. It is therefore necessary to pass the gas through a preparation unit before using it for carburizing. The preparation unit is, of course, an



Rotary hearth hardening furnace for crown wheels and bevel pinions

essential part of the Wild-Barfield prepared town's gas carburizing equipment. In addition to the preparation unit, the Rover equipment also includes a Holmes-Maxted catalytic sulphur removal plant, through which the gas from the mains is passed before is passes through the preparation unit. The sulphur removal plant reduces the concentration of organic sulphur compounds from an average of 25 grammes to three grammes per 100 cubic feet.

From the sulphur removal plant, the gas is fed through an oil filter and a flow-meter to a special catalyst furnace. Twin thermo-couples are fitted to the catalyst furnace, one registering the temperature of the heating elements, 960 deg C, and the other the temperature of the catalyst retort, 925-935 deg C. From the catalyst retort, the gas passes through an air cooler and thence to a silica gel drier. The average composition of the gas before passage through the preparation plant is:—

Carbon dioxide	3·7 per cent
Heavy hydrocarbons	2·8 per cent
Oxygen	0·9 per cent
Carbon monoxide	22·5 per cent
Hydrogen	46·4 per cent
Methane	15·1 per cent
Nitrogen	8·6 per cent

Of these constituents, the carbon dioxide, heavy hydrocarbons and oxygen are deleterious so far as carburizing is concerned, therefore the function of the preparation plant is to remove them. The percentages of these constituents after preparation are:—

Carbon dioxide	0·2
Heavy hydrocarbons	1·35
Oxygen	Nil

Because of the importance of the gas composition, equipment for checking it has been installed in the carburizing section. One check is carried out in the morning and one in the afternoon.

The carburizing furnace has a chamber 36 in diameter by 78 in deep to take a retort 24 in diameter by 48 in deep. Considerable experimentation was carried out before the actual carburizing conditions were established. Incidentally, there are differences between the treatment for crown wheels and that for pinions.

As can be seen from one of the illustrations, the crown wheels are loaded on special charge carriers; 75 wheels make up one charge. The charge is then loaded into a retort, which is purged to expel all oxygen before the retort and its load are transferred to the carburizing furnace. Carburizing is effected at 940 deg C, but the introduction of the cold retort into the furnace reduces the furnace temperature to about 700 deg C. A period of from one to one and three-quarter hours is necessary to bring the furnace up to the control temperature. During this period the prepared gas is introduced to the furnace at the rate of 70 cubic feet per hour. When the control temperature is reached, the gas flow is increased to 240 cubic feet per hour and active carburization is carried on for 2¼ hours.

During the active carburization period, carbon is fed into the steel as fast as it can be absorbed. At the end of the active carburization period, there is a diffusion period to allow the absorbed carbon to be diffused into the steel. Normal practice for the diffusion period is to close the gas inlet and outlet valves to allow the gas in the retort to remain static. For crown wheel treatment, The Rover Co. Ltd. have found it advisable to maintain a flow of gas at 50 cubic feet per hour throughout the diffusion period.

A uniform distribution of gas throughout the charge is essential. For crown wheels this is obtained by means of eight distribution tubes round the walls of the retort and a central distribution vent. Three test pieces, one at the top, one midway and one at the bottom, are included with every charge. The radial positions of the test pieces are varied for each charge; that is, in one charge, the top test piece may be to one side of the retort, while the bottom one is at the other side and the centre one is in the middle; with the succeeding charge, the test pieces may be arranged with the top and bottom towards the middle and the central one to one side.

At the end of the one hour diffusion period, the retort is removed from the carburizing chamber to a cooling chamber. Finally the retort is taken to the unloading pit, but unloading is not effected until the temperature has fallen below 100 deg. As soon as the charge is unloaded, the test pieces are delivered to the metallurgical laboratory for complete tests before any further treatment is carried out on the crown wheels comprising the charge.

Bevel pinions are treated at the rate of 180 per charge. The method of loading the carrier can be seen in one of the illustrations. As with the crown wheels, test pieces are included with every charge. After being charged, the retort is purged and is then transferred to the carburizing chamber where the period to bring the work up to the control temperature, 940 deg C, is the same as for the crown wheels. Active carburizing extends over a period of 2½ hours, against the 2¾ hours for the crown wheels, and the gas flow is at 240 cubic feet per hour. The diffusion period lasts one hour, but for the bevel pinions the gas flow ceases at the end of the active carburization period. Another difference between the methods for crown wheels and the bevel pinions is that for the pinions, gas distribution is completely effected by the eight tubes round the wall of the retort, the centre vent being stopped-off.

Hardening is effected in a Gibbons, Wild-Barfield rotary hearth furnace. Incidentally, this furnace does not have a controlled atmosphere. Passage through the furnace takes 43 to 45 minutes, in which period the work is brought to 800-810 deg C. The crown wheels are quenched in a Gleason press, with Vaughan's No. 2 oil as the quenching medium. The temperature of the quenching medium is controlled to 55-65 deg C. Pinions are tank quenched in the same oil maintained at the same temperature. To complete the treatment the components are tempered at 160 deg C for one hour and are then shot peened. A nominal case depth of 0·040 in is specified with a maximum of 0·045 in and a minimum of 0·037 in. Specified Rockwell hardness on the C scale is 58-60 for crown wheels and 60-62 for the bevel pinions.

There is general agreement that metallurgically the product from gas carburizing is of a more uniform and a higher quality than was obtainable when pack carburizing was employed.

For components such as crown wheels and bevel pinions, The Rover Co. Ltd. consider this question of quality to be the sole criterion. But with the improved quality that is possible with the very close control of conditions that is a feature of gas carburizing, there have also been substantial production economies. For example, the labour charges for loading and luting pack-hardening boxes are very much greater than those for loading the carriers for gas carburizing. A further advantage is that the time required to produce a given case is very much less with gas carburizing. When pack hardening is employed the actual carburizing time is in the order of seven hours against $2\frac{1}{2}$ hours for gas carburizing. There is, therefore, a decided advantage in fuel costs, and a given output can be handled in a smaller plant. It is also almost certain that the charges for the component carriers for gas carburizing will be appreciably less than the charges for hardening boxes for the same volume of work.

A HYDRAULIC CLUTCH CONTROL

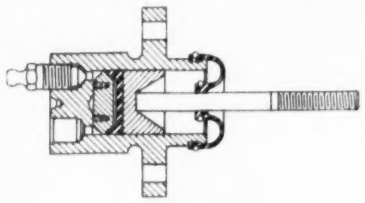
The New Lockheed Unit

To overcome some of the disadvantages associated with mechanical clutch actuation, the Lockheed Hydraulic Brake Co. Ltd., of Leamington Spa, have introduced a new hydraulic clutch operating unit with a flexible pipe as the only connection between the pedal and the clutch. Of recent years, much progress has been made in the reduction of noise in motor vehicles. As a result, rattles emanating from components such as the clutch controls are all the more annoying because of the absence of other noises to mask them. In addition, these controls have an increased tendency to rattle owing to the greater flexibility of modern engine mountings. Moreover, unless some form of positive tie or a compensating device is incorporated to prevent or otherwise obviate the effects of longitudinal motion of the engine relative to the frame, trouble may be experienced with clutch judder when mechanical controls are fitted.

With the new Lockheed control, the clutch pedal operates a hydraulic master cylinder which, together with the brake master cylinder, is incorporated in a single casting. Pressure on the pedal forces the fluid through the flexible pipe to a small slave cylinder that operates the clutch release mechanism. The very low friction of the system and the adoption of suitable piston areas has resulted in a light and positive clutch pedal operation. Maintenance is reduced to a minimum, since the hydraulic system is completely enclosed and self-lubricated, so that there are no lubrication points in inaccessible parts of the chassis.

In the casting that forms the body of the master cylinder unit, the lower part of the hydraulic reservoir is divided into two compartments. Thus, in the event of a leak occurring in either the brake or the clutch system, there is still enough fluid left for operation of the unaffected circuit. This reservoir, which is immediately above the two cylinders, is served by a filler cap incorporated in the cast aluminium cover that is secured on top of it by four $\frac{1}{4}$ in diameter set screws.

The plunger is a running fit in the cylinder, and is about 2 in long. Round it is an annular space approximately $1\frac{3}{8}$ in long. A seal is retained in position by an integral collar, near the outer end of the plunger, in the base of this space. A $\frac{3}{16}$ in diameter hole drilled through the cylinder wall from the

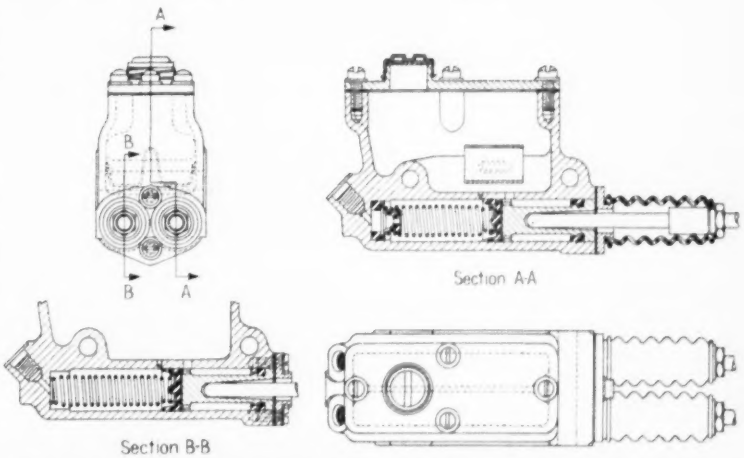


The clutch unit

reservoir is at all times in communication with this space for bleeding purposes. This arrangement has the additional advantage that the fluid seal at the rear does not work under pressure and there are practically no wear or leakage problems.

The cup washer is held against the head of the plunger by a compression spring seating in a dished steel disc; the other end of the spring bears against the end of the cylinder casting. When the clutch pedal is released, the cup washer is just clear of a small recuperator port that communicates between the working part of the cylinder and the reservoir. Not only does this permit the return of oil which may have leaked past the plunger cup washer, but it also allows for expansion and contraction. Holes for bleeding are drilled parallel to the axis of the plunger between its front face and the annular groove. The brake master cylinder arrangement is similar to that of the clutch operating cylinder, but a check valve is incorporated in the outlet to ensure that fluid cannot drain back from the brake pipe lines through the cylinder to the reservoir during the bleeding operation.

A single pipe line connects the master and slave cylinders. The union to which this pipe is fitted is screwed into the lower tapped hole in the slave cylinder. As can be seen in the illustration, the upper tapped hole carries a bleed screw. The cylinder casting is flanged for two $\frac{3}{16}$ in diameter bolts by means of which it is mounted on the chassis. The plunger is a running fit in the cylinder, but it is only about $\frac{1}{4}$ in long. A cup washer seats on the inner face of the plunger. When in operation, it is retained in position by hydraulic pressure, and when released, by a spring loaded stop between it and the closed end of the cylinder. The return action is, of course, effected by the clutch springs.



The clutch and brake cylinders are incorporated side by side in the one master unit

STRUCTURAL DESIGN

Part II: The Front End Structure

By T. K. Garrett, A.M.I.Mech.E., A.F.R.Ae.S.

Continued from page 111.

IN the earlier article entitled "Automobile Dynamic Loads", two cases are given that are likely to be critical at the front end. They are Case 1, hitting a bump while braking, and Case 2, cornering. At the front $R = \frac{3904}{2\times4\cdot5} =$ 434 lb. The loads for these two cases are:

Case 1

Vertical $= 1\cdot5\left\{3R\left(1+\frac{\bar{y}}{2B}\tan\theta\right)+\frac{W\bar{y}}{2B}\right\}$

$= 1\cdot5\left\{3\times434\left(1+\frac{27}{2\times77}\times0\cdot743\right)+\frac{2060\times27}{2\times77}\right\}$

$= 2750$ lb

Rearward $= 1\cdot5\left\{R(3\tan\theta+1)+\frac{W\bar{y}}{2B}\right\}$

$= 1\cdot5\left\{434(3\times0\cdot743+1)+\frac{2060\times27}{2\times77}\right\}$

$= 2640$ lb

Lateral $= 0$.

Case 2

Vertical $= 1\cdot5W\left\{\frac{\text{Cos }\psi}{\sin\phi}+\mu\cos\psi\frac{\bar{y}}{2B}\right\}$

$= 1\cdot5\times2060\left\{\frac{0\cdot208}{0\cdot407}+0\cdot208\times\frac{27}{2\times77}\right\}$

$= 1690$ lb

Rearward $= 1\cdot5\ \mu W\cos\psi$

$= 1\cdot5\times2060\times0\cdot208$

$= 642$ lb

Lateral $= 1\cdot5\mu W\frac{\cos\psi}{\tan\phi}$

$= 1\cdot5\times2060\times\frac{0\cdot208}{0\cdot445}$

$= 1440$ lb.

These loads are summarized in Fig. 20. It must be noted that the effect of brake torque in Case 1 is not included in that diagram; however, it must be taken into account and the brake manufacturers should be consulted to find out what is the maximum possible torque. For the purpose of this article, it will be calculated from the following considerations:—

Retardation force due to braking $= \mu g$

Therefore $\mu = 1\cdot0$

Front end vertical reaction under braking conditions $= 4\cdot5+\frac{W\bar{y}}{2B}$

$= 797$ lb/wheel

Since $\mu = 1\cdot0$, the drag load $= 797$ lb

Tyre rolling radius $= 11\cdot25$ in

Brake torque $= 795\times11\cdot25$

$= 8950$ lb-in.

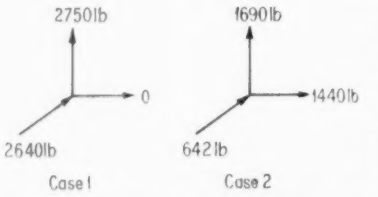


Fig. 20. The loads applied at each front wheel

With the brake torque treated separately in this manner, all the drag loads, as well as the vertical loads, may be considered as acting through the wheel centre. Only the side loads are acting at the point of contact between the tyre and the ground.

Next, the loads to the four wishbone bearings must be estimated from the geometry of the front suspension system, Fig. 21. For the following calculations it is assumed that the upper wishbone bearings are spaced 5 in apart, and the distance between the lower pair is 8½ in.

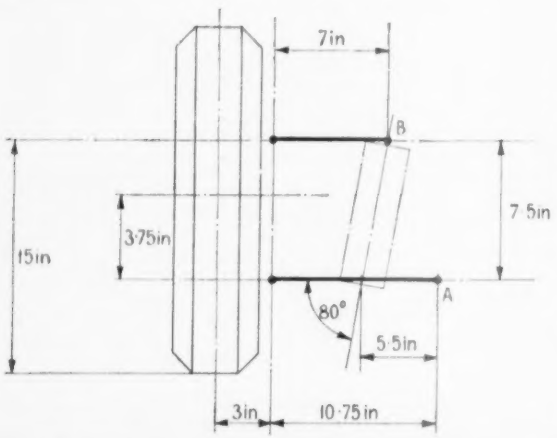


Fig. 21. The geometry of the front suspension system

Case 1

Loads in the transverse vertical plane

Moments about A for vertical component of spring and bump stop reaction

$= \frac{2750\times10\cdot75}{5\cdot5} = 5370$ lb

Horizontal component

$= \frac{5370}{\sin 80}\times\cos 80 = 946$ lb reacted at A

Moments about outer bearing for inner bearing vertical reaction

$= \frac{5370\times5\cdot25}{10\cdot75} = 2620$ lb/pair

Differential horizontal loads at outer bearings due to offset of wheel centre

$= \frac{2750\times3}{7\cdot5} = 1100$ lb.

Drag Loads

It will be assumed that these loads may be applied with the wheel either at full bump when the upper wishbone arm bearing will take most, or at normal laden position in which a large proportion will be reacted by the lower wishbone. At full bump, the proportion of the drag load reacted by the top bearings is $6\cdot75/7\cdot5\times2640 = 2480$ lb/pair, and the differential load due to the offset of the wheel from the inner bearing is $(2480\times10)/5 = 4960$ lb. At the normal laden position the proportion to the lower bearings is $3\cdot75/7\cdot5\times2640 = 1320$ lb/pair, and the differential load is $1320\times13\cdot75/8\cdot5 = 2140$ lb.

Brake torque

Differential load due to brake torque on outer wishbone bearings

$= \frac{8950}{7\cdot5} = 1200$ lb

Differential loads to upper wishbone inner bearings

$= \frac{1200\times10}{5} = 2400$ lb

Differential load to lower wishbone inner bearings

$$= \frac{1200 \times 13{\cdot}75}{8{\cdot}5} = 1940 \text{ lb.}$$

It is now necessary to sum the loads at each inner bearing and to check that they balance with the applied loads at the wheel. Loads directed inwards, upwards, and to the rear will be considered as positive.

Top rear bearing:
- Vertical component = 0
- Lateral component = 550 + 4960—2400 = + 3110 lb
- Rearward component = + 1240 lb

Top front bearing:
- Vertical component = 0
- Lateral component = 550—4960 + 2400 = —2010 lb
- Rearward component = + 1240 lb

Lower rear bearing:
- Vertical component = —1400 lb
- Lateral component = —473—550 + 2140 + 1940 = + 3047 lb
- Rearward component = + 660 lb

Lower front bearing:
- Vertical component = —1400 lb
- Lateral component = —(473 + 550 + 2140 + 1940) = —5103 lb
- Rearward component = + 660 lb

Check on balance against the loads in Fig. 20 :

Vertical = —2620 + 5370 = 2750 lb (error zero)

Lateral = + 3110—2010 + 3047—5103 + 946 = 10 lb (error 0·1%).

The rearward components will not balance because of the two different assumptions made with regard to the vertical position of the wheel.

Case 2

Loads in the transverse vertical plane

Moments about A for vertical component of the spring and bump stop reactions

$$= \frac{1690 \times 10{\cdot}75}{5{\cdot}5} = 3300 \text{ lb}$$

Horizontal component

$$= \frac{3300}{\tan 80} = 582 \text{ lb}$$

Moments about outer bearing

$$= \frac{3300 \times 5{\cdot}25}{10{\cdot}75} = 1610 \text{ lb.}$$

Differential horizontal loads at outer bearings due to offset of wheel centre

$$= \frac{1690 \times 3}{7{\cdot}5} = 677 \text{ lb.}$$

Drag loads

At full bump the proportion of the drag load reacted by the top bearings is 6·75/7·5 × 642 = 578 lb/pair, and the differential load due to the offset of the wheel from the inner bearing is (578 × 10)/5 = 1156 lb. At the normal laden position, the proportion reacted by the lower bearings is 3·75/7·5 × 642 = 321 lb/pair, and the differential load is (321 × 13·75)/8·5 = 519 lb.

Side loads

Moments about lower bearing for upper bearing reactions

$$= \frac{1440 \times 7{\cdot}5}{7{\cdot}5} = 1440 \text{ lb}$$

Load to each inner bearing

$$= \frac{1440}{2} = 720 \text{ lb}$$

Moments about upper bearing for lower bearing reaction

$$= \frac{1440 \times 15}{7{\cdot}5} = 2880 \text{ lb}$$

Loads to each inner bearing

$$= \frac{2880}{2} = 1440 \text{ lb.}$$

In all these calculations, it has been assumed that the wishbone links are all horizontal. This assumption has been made to avoid further complication of the calculations, and to make them easier to follow. In practice allowance must be made for angularity of the links.

Summing the loads, we have :

Top rear bearing:
- Vertical component = 0
- Lateral component = 338 + 1156—720 = + 774 lb
- Rearward component = + 289 lb

Top front bearing:
- Vertical component = 0
- Lateral component = 338—1156—720 = —1538 lb
- Rearward component = + 289 lb

Lower rear bearing:
- Vertical component = — 805 lb
- Lateral component = —291—338 + 519 + 1440 = + 1330 lb
- Rearward component = + 160 lb

Lower front bearing:
- Vertical component = — 805 lb
- Lateral component = —291—338—519 + 1440 = + 292 lb
- Rearward component = + 160 lb

Checking the balance against the loads in Fig. 20, we have:

Vertical = —1610 + 3300 = 1690 lb (error zero)

Lateral = 774—1538 + 1330 + 292 + 582 = 1440 lb (error zero)

Rearward — No balance possible.

A word of warning is necessary in connection with balancing : it is easy to confuse loads with reactions. For that reason, when drawing diagrams it is best always to use red pencil to mark arrows representing reactions and blue pencil for arrows representing loads. It will be noticed that the lateral loads are balanced by including the lateral component of the spring and bump stop load. At the top end, this load is taken by the structure and not by the upper pair of bearings. There is one more effect to be superimposed on the system of loads, and that is the steering rod reactions.

This may be done as follows :

Case 1
- Drag load = 2640 lb
- Moment about swivel pin = 2640 × 2·4 = 6340 lb-in
- Effective length of steering arm = 6·375 in
- Load to steering arm = 6340/6·375 = 995 lb.

The steering rod reaction will apply outwardly directed loads at the four wishbone bearings ; its effect will be distributed between the top and bottom pair in proportion to the vertical disposition of the steering rod relative to the two pairs of bearings. Moreover, if the rod is not horizontal there will be a vertical component to its reaction which will be balanced mainly between the road wheel and the steering box. By virtue of the geometry of the system only a very small proportion of this vertical component will be reacted through the wishbone bearings. There may also be a fore and aft component.

Case 2 can be calculated in a similar manner, but the wheel will be on lock, with the result that the effective length of the steering arm will be different. Since this article is only concerned with showing the principle of tracing the path of the loads through the structure, the steering reactions will be ignored. This will avoid undue complications of the calculations. The loads taken to be correct for Cases 1 and 2 are therefore those summarized in Figs. 22 and 23.

When the loads have been calculated, the first step is to check the bearings and their attachment to the structure in the manner already described for the rear end. Next, the path of the loads must be traced through the structure. The shock absorber, which carries the upper wishbone bearings, is mounted on a platform housed in a vertical channel and although Figs. 22 and 23 show no vertical components on the top bearings, the shock absorber will introduce vertical loads and moments. These may be determined by experiment with the particular shock absorber to be used, or by consultation with the manufacturer.

There is no very accurate method of estimating the stresses in flat plates, and loading normal to the plane of a plate should be avoided where possible, owing to the danger of fatigue failures arising from high local bending stresses. Attachment bolts securing the shock absorber base to the platform should be as close as practicable to the vertical walls of the channel in which it is housed, and the pressed steel spring-seat should be shaped in such a way as to reinforce these attachment points.

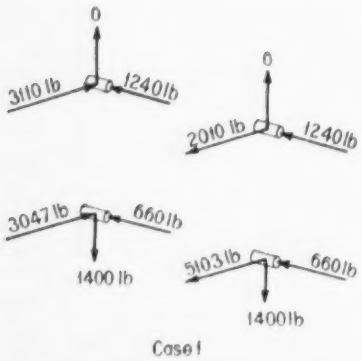


Fig. 22. The loads applied at the inner wishbone bearings when the wheel hits a bump with the brakes full on

Ideally, the bolts ought to be backed up by channel sections welded to the underside of the platform in such a manner that they carry the loads in bending to the sides. However, in this case the spring takes up too much space for this arrangement to be possible. An approximation of the stresses may be made by assuming that the load is taken to the vertical walls by the flat plate in bending and then calculating the stress by a method similar to that employed for checking the side load stresses in the rear spring attachment bracket.

So far as the shear attachment between the platform and the channel is concerned, Case 1 will be critical. The direct lateral shear is 3110—2010 = 1100 lb, while the couple, about a vertical axis through the centre of the spindle, due to the differential loads at the bearings is 3110×2·5 + 2010×2·5 = 12790 lb-in. This couple must be reacted by shear in the spot welds to the sides of the vertical channel. These sides are 8½ in apart so that the shear per side is 12790/8·5 = 1500 lb, add the direct shear of 1100/2 + 1500 = 2050 lb.

Superimposed on this is the effect of the drag load of 2480 lb acting along a line 8½ in from the inner wall of the 18 s.w.g. channel. The direct shear will have to be taken to the inner wall by at least 2480/375 = 7 spot welds, while the moment due to the offset must be taken by the spot welds on the side walls of the channel. This load, of 2480×8·5/8·5, added to the 2050 lb gives a total of 4530 lb, and at least 13 spot welds are needed. Although the vertical load will be relatively small, it may nevertheless be necessary to incorporate additional spot welds to carry it in shear to the walls of the channel.

The channel itself must be checked as a beam loaded in two planes. That is to say, its modulus of section about its two major axes must be calculated, and the bending stresses estimated and added together at the appropriate points. The effective line of action of the horizontal loads is in the plane indicated by the chain dotted line in the upper diagram of Fig. 24a. These loads are the drag of 2480 lb, and inwardly directed load of 4530 lb on the rear wall of the channel and an outwardly directed one of 150—550 + 2480 = 2000 lb on the front wall. From the principle of moments, it follows that at the top of the channel the shear reactions are $\frac{7}{16}$ of the total loads, while at the base they are $\frac{9}{16}$ of them.

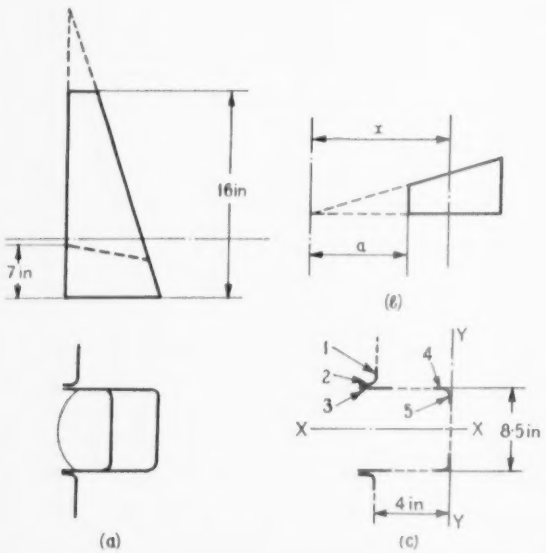


Fig. 24. A diagrammatic illustration of the vertical channel carrying the front suspension loads

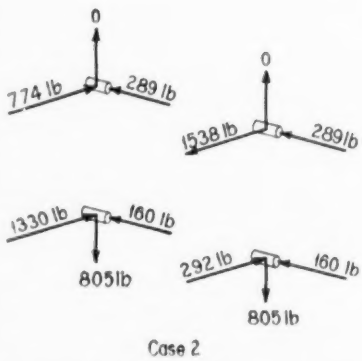


Fig. 23. The loads applied at the inner wishbone bearings when the vehicle is cornering at a speed just below that at which skidding occurs

The greatest bending moments will occur at the point of application of the loads, 9 in below the top of the channel. At this point, the section is as shown in Fig. 24c, in which the unstable portions of the section are shown dotted. Because it is symmetrical about the XX axis, its moment of inertia may be approximated as follows:

1. $Ad^2 = (16\times0{\cdot}036\times0{\cdot}036)\times4{\cdot}53^2$ = 4·33 in⁴
2. $Ad^2 = (16\times0{\cdot}036\times0{\cdot}036)\times4{\cdot}25^2$ = 3·74 in⁴
3. $Ad^2 = (32\times0{\cdot}048\times0{\cdot}048)\times4{\cdot}25^2$ = 13·30 in⁴
4. $Ad^2 = (16\times0{\cdot}048\times0{\cdot}048)\times4{\cdot}25^2$ = 6·65 in⁴
5. $Ad^2 = (16\times0{\cdot}048\times0{\cdot}048)\times3{\cdot}87^2$ = 5·50 in⁴

Total 33·52 in⁴

This total represents only half the section, so that the total moment of inertia is 67·04 in⁴, and Z = 67·4/9·07 = 7·4 in³

Bending moment $\frac{7}{16}\times2480\times9$ = 9740 lb-in

Bending stress $\frac{M}{Z} = \frac{9740}{7{\cdot}4\times2240}$ = 0·59 ton/in².

The shear stress is 2480/8·5×0·048 = 6080 lb/in². This will be satisfactory.

About the other axis, the moment of inertia must be calculated by the tabular method. Each wall of the channel is loaded differently, but only the heaviest loading need be considered, because practical considerations require that the section shall be symmetrical.

Item	Size	Area	x	x²	Ax	Ax²
1.	0·036×0·57	0·0205	4·0	16·0	0·082	0·328
2.	0·036×0·57	0·0205	4·28	18·3	0·088	0·375
3.	0·048×1·54	0·0739	3·8	14·4	0·281	1·065
4.	0·048×0·77	0·0369	0·38	0·144	0·014	0·005
5.	0·048×0·77	0·0369	—	—	—	—
		ΣA = 0·1887			Σ = 0·465	1·773

In this case, the local moments of inertia are small by comparison with the total, and may be ignored.

$\bar{x}_B$ = 0·465/0·1887 = 2·47 in

$\bar{x}_T$ = 4·57—2·47 = 2·1 in

I = 1·773—0·1887×2·47² = 0·62 in⁴

Z_B = 0·62/2·47 = 0·25 in³

The bending moment is $\frac{7}{16}\times4530\times9$ = 17800 lb-in

$\frac{M}{Z} = \frac{17800}{0{\cdot}25\times2240}$ = 31·9 ton/in².

To this must be added the stress due to bending about the XX axis, and the total equals 32·5 ton/in². In addition, the vertical components from the lower wishbone bearings must be taken into consideration.

The axes of the bearings are $\frac{5}{16}$ in, measured horizontally, from the inner wall of the channel and 5 $\frac{9}{16}$ in from its outer edges. It will be safest, therefore, to assume that all the vertical load is taken by the inner wall together with about $\frac{1}{8}$ in of each outer wall of the channel. At both bearings the

load is 1400 lb downwards, so that the stress is: $1400/0{\cdot}048\times(4{\cdot}25+0{\cdot}625)\times2240=2{\cdot}7$ ton in^2. This brings the total to 35·2 ton in^2. This stress is too high, so the channel must be strengthened. This can can be done by increasing either the depth of the channel, or the gauge of the material, or a doubling plate may be spot welded in. Increasing the depth is preferable, both from the point of view of ease of production and because the strength increases as the square of the depth. It is, of course, most essential to ensure also that there is an adequate path to carry the loads from the bearing lugs to the vertical channel.

Shear at the top end is likely to be the critical factor for the side walls. The heaviest load is on the rear wall, and the reaction at the top is $4530\times\frac{7}{16}=1980$ lb. It will be advisable to use gas welding to attach the top of the channel to the top of the wheelarch, because the length of the joint at that point is only $1\frac{1}{4}$ in. The shear stress in the wall is $1980/1{\cdot}25\times0{\cdot}048\times2240=14{\cdot}8$ ton in^2. This also is too high, and it would be better to limit the stress to 9 ton in^2 by increasing the depth of the section. The inner wall is obviously satisfactory.

The lateral and longitudinal loads at the top end of the channel are reacted by the top of the wheelarch, which acts as a beam supported by the front bulkhead, or radiator support, and the dash. Although this beam is curved, it cannot distort under normal loads because it is stabilized at its inner edge by the wheelarch inner panel, and at the outer edge by the vertical side member which passes over it. Checking the stresses in it is a relatively simple matter and will not be elaborated on.

At the lower end of the channel, the reactions are $4530\times\frac{9}{16}=2550$ lb at the rear, and $2000\times\frac{9}{16}=1130$ lb at the front. The drag load reacted at the bottom is $2480\times\frac{9}{16}=1400$ lb. All these loads together with the horizontal loads from the lower wishbone bearings are reacted by the portal frame.

The portal frame

The portal frame is a horizontal component built up from two side members overhung from the under side of the front floor, and one cross member at the extreme front end. All these members are 6 in wide by 16 s.w.g., and have 1 in flanges at each side. They are intended to take only loads in the horizontal plane, since the vertical loads at the front are catered for by the wheelarches, suspension channels, front cross member or bulkhead, and dash.

The total loads on the portal frame are $1130+5103=6233$ lb outwards at the front wishbone bearing, and $2550+3047=5597$ lb inwards at the rear bearing. In addition there is a couple of $1320\times8{\cdot}06=10600$ lb-in due to the offset of the drag load from the neutral axis of the side member. Fig. 25 and Fig. 26 give a summary of these loads for the design case 1, which is hitting a bump when braking. It is reasonable to assume that the bump may be struck either by one wheel or by two front wheels simultaneously. The illustrations show both conditions diagrammatically; Fig. 25 on the very severe assumption that there is zero load on one side, and Fig. 26 on the assumption that the load is equal on both sides.

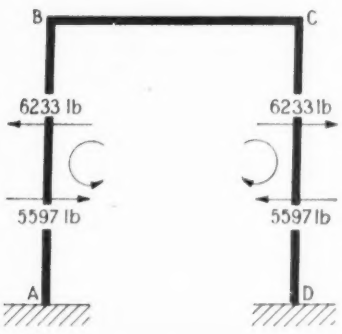


Fig. 26. The loads on the portal frame under symmetric loading conditions

There are several ways of estimating bending moments on portal frames. One of the best is the Hardy Cross method described in great detail in "Notes on Some Modern Methods of Structural Analysis" by F. Simpson, published by the Draughtsman Publishing Co. Ltd. This method appears to be more widely known in the U.S.A. than in this country. It has many advantages over the other methods for the solution of continuous beam problems, and the reader is advised to adopt it. The bending moment diagrams will be similar to those in Figs. 27 and 28. In these the base line is drawn to a scale representing the true dimensions of the frame so that the points A, B, C and D in the diagrams correspond respectively to the points A, B, C and D of the frame, Figs. 25 and 26.

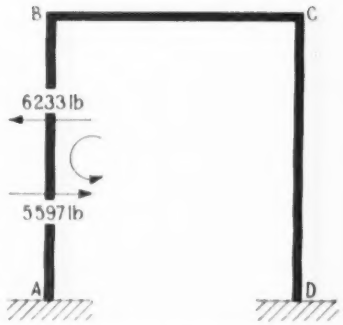


Fig. 25. The loads on the portal frame under asymmetric loading conditions

When the bending moments have been calculated, the shear reactions at the base of the frame and at the cross member may be determined. Then the checking of the shear attachments, bending and direct stresses, and shear stresses in the members is a fairly simple matter. Care must be taken to check not only points where the loads are greatest, but also where the section is weakened by cut-outs incorporated to provide clearance for pedals, etc.

It must always be remembered that shear loads induce end loads at the edges of holes in the webs of beams and in the booms. This means that the stresses in the booms must be calculated in the normal manner but, if the dimensions of the hole are large by comparison with the depth of the beam, the additional local stresses will be large and must be superimposed on the normal stresses. These local

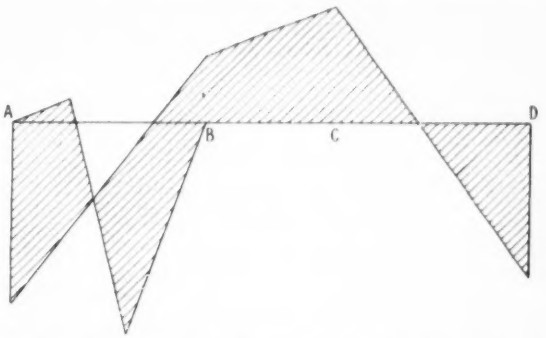


Fig. 27. Bending moment diagram for the portal frame under asymmetric loading

stresses are a result of shear deflection, shown exaggerated in Fig. 29b, which imposes additional bending loads at the sections, such as BB in Fig. 29a, adjacent to the corners of the hole.

In the case illustrated, the best method of calculating the stresses is to divide the bending moment at the worst section, which will be assumed to be BB, by the depth between the neutral axes of the two sections MN and OP. This will give the end loads, on the two sections. Then the stresses may be obtained by dividing the loads by the cross sectional area. The local bending stresses are calculated in the following manner.

The point of contraflexure in the case of a symmetrical hole will be mid way between its two ends. If the hole is not symmetrical the point may be estimated, by calculus methods, from the fact that the stiffness of a beam is proportional to EI/L, where E is Young's Modulus, I is the moment of inertia, and L is the length of the beam.

In Fig. 28, contraflexure takes place on section AA. It is assumed that the shear "S" is shared between the sections ST and UV in proportion to their depth, so that in this

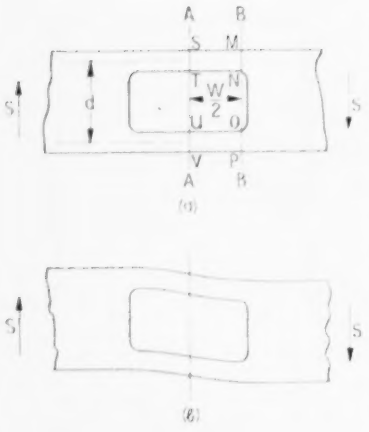


Fig. 29a. A beam with a large rectangular hole in its web

b. The type of deflection experienced at the hole

case S/2 lb is taken by each. This shear will give rise to a local bending moment at sections MN and OP of S/2×W/2, where W is the width of the hole. The stresses induced by those bending moments may be determined by dividing by the modulus of the sections. These stresses must then be superimposed on the P/A stress, and the total must not be greater than the strength of the section.

The dash and toe board structure

An analysis has now been made of the way in which the horizontal loads at the front are taken in the portal frame, and the vertical ones into the suspension channel. Two more checks must be made. The first is to ensure that there is an adequate path by which vertical loads may pass back to the dash. Secondly, the support afforded at the root of the portal frame must be checked.

There are two ways in which the vertical loads may be carried back from the suspension channel to the dash. The first is cantilever fashion along the rear portion of the inner vertical wall of the wheel arch, and the second is in the manner of a simply supported beam, the beam being the whole of the inner vertical wall of the wheel arch. In this second case, the supports for the beam are the toe board at the rear and

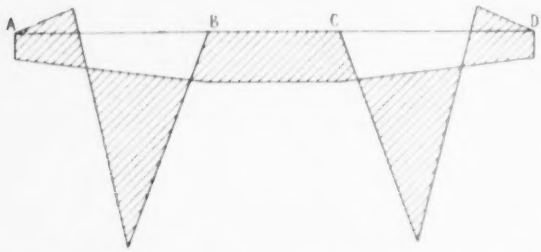


Fig. 28. Bending moment diagram for the portal frame under symmetric loading

the bulkhead that supports the radiator at the front. The proportion of the load that is carried forward is transferred from the bulkhead to the rear again by the extension, over the wheel arch, of the dash side. This extension is a cantilever beam approximately 9 in deep. It is stabilized at its top edge by its attachment to the drip channel at the edge of the bonnet opening, and at its bottom edge by the top of the wheel arch to which it is spot welded.

In practice, the path that the loads actually take will be determined by the relative stiffness of the two systems; part of the load will be carried by one path and part by the other. All the beams involved are relatively deep, so the stresses will not be very great. In this case, it will be best to check the structure on each of the two assumptions mentioned. The determination of the reactions at the toe board and the front bulkhead and of the stresses in the beams is a simple matter and will not be elaborated upon here.

Next the toe board must be checked. The loads applied to this component are the vertical shear from the inner face of the rear portion of the wheel arch and lateral shear loads from the root end of the portal frame. Although the portal frame is spot welded to the floor, the support thus afforded is negligible because of the break caused by the propeller shaft tunnel. It follows that support for these lateral loads must be furnished by the toe board.

At first sight, because of its size, the toe board would appear to have adequate strength to withstand any loads likely to be applied to it. However, there is one feature that may cause trouble. The cut out in the lower edge where it clears the gearbox may be likened to a notch in a more conventional type of beam; it induces stress concentrations because of the discontinuity of structure that it represents. There are several ways in which the toe board may be strengthened to carry the loads across this discontinuity. One of these is shown in Fig. 30, in which A B C D represents the toe board. Vertical, top hat section members are spot welded above E and F to the front face of the toe board, and a horizontal member, PQ, is similarly attached, to the rear face of the toe board, immediately above the cut out for the gearbox.

It may be necessary to modify the shape of PQ in order to clear cut outs for pedals, etc. This can be done by bending the horizontal member at the points where it crosses the two vertical top hat sections, so that its ends P and Q may be either higher or lower than shown in Fig. 29. A less expensive method of reinforcing the toe board would be to press vertical swages into it at the positions EX and FY instead of using the top hat sections. It might even be possible to substitute a horizontal swage for the member PQ, but this would have to be discontinuous where it crosses the vertical swages. Such an arrangement would only be practicable when the loading is relatively light, since there would be a tendency for stress concentrations to occur around the points of discontinuity on the horizontal swage.

The way in which this structure carries the loads applied to it is as follows. The vertical shear from the wheel arch is reacted by the toe board acting as a simple beam supported at each side, AD and BC. At the lower edges DE and FC, the lateral shear loads from the portal frame are taken by the cantilever beam formed between DE and RP. The bending moment is the shear S multiplied by the vertical distance between the lower edge and the horizontal member, and the end loads, induced at P and R in PD and RE respectively, are the bending moments divided by the distance between P and R. These loads are then diffused in shear from the remaining portions of the vertical members into the panel above the horizontal member. In the asymmetrical case, the moment will be taken off by reactions at AP and BQ, while in the symmetrical case, an equal and opposite moment will be induced by shear on FC, so that the system will be self-balanced.

General remarks

The calculations necessary to check the strength of a complete vehicle structure take approximately 3 months to complete, and occupy much more space than that available for this article. Therefore, only a selection of problems fairly representative of all those likely to be encountered have been presented here. Moreover, in order to shorten the work, many simple approximations have been made. In many of the examples given, these approximations would have been justified as a means of speeding up the work in the design office, because the stresses were obviously appreciably lower than those that the component under examination was capable of supporting. Thus there was a fair margin of safety to cover any errors which might arise as a result of the approximations. More accurate methods must be used if the reserve factor is small.

A reserve factor is defined as the ratio between the stress that the component is

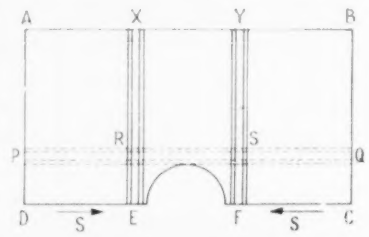


Fig. 30. A diagrammatic illustration of the toe board structure

capable of carrying and the stress likely to be experienced. When recording the calculations it is advisable to quote the results in the following manner:

Stress = . . . ton/in² @ . . . ton/in² R.F. = . . .

The abbreviation R.F., of course, refers to the reserve factor, and it is convenient to have it in the record in order that, at a later date, a quick assessment may be made of the advisability or otherwise of incorporating modifications that will reduce the strength of the component. Such modifications are often called for by the production departments to facilitate assembly, etc.

It is often thought that stress analysis is work for mathematicians. This is not so. The standard of mathematics required is not very high, but the practical engineering knowledge that must be applied calls for particularly sound mechanical sense and experience. However, a knowledge of more advanced mathematics, although not essential, is useful because it can often be applied to obtain more accurate solutions to problems the answers to which must otherwise be approximated.

Careful records should be kept of the calculations, and they should be compared with results from tests with strain gauges. In this way false assumptions may be detected and experience accumulated so that in the course of time more and more accurate forecasts may be made by mathematical methods of the stresses that will be experienced in vehicle structures. Moreover, an accumulation of the results of mechanical tests of standard components, spot welds, etc., will not only improve the accuracy of the estimates of strength of structures, but curves giving design data will also enable the work to be done more rapidly. With practice, these analytical methods can be improved until extremely reliable results are consistently obtained so that a great deal of development work is saved.

CERAMETALLIC BRAKE LININGS

A New Friction Material for Aircraft Brake Linings

A NEW brake lining material has been developed recently in the United States of America. It is now in production in the Bendix Products Division plant. Some time ago, it became apparent that if the braking requirements of modern aircraft were to be met satisfactorily, a new type of brake lining would have to be found. This called for an entirely new approach, eliminating resin and substituting a base material that would be stable under the highest temperatures. Although the requirements for aircraft brakes are different from, and in most respects more severe than those for motor vehicles, it would appear likely that a new material of this type might be used to solve the problems encountered with modern motor vehicle brakes. These problems arise, of course, because of the almost total enclosure of the wheels within the body shell, and because of the relatively high speeds of which the present-day cars are capable. That the new brake linings are suitable for very heavy duties is indicated by the stipulation that a certain aircraft weighing 17 tons must be stopped in 20 seconds from a ground speed of 130 m.p.h.

After extensive research, the Bendix engineers developed the new material which is a combination of ceramic and metallic ingredients; it is called Cerametallic. The ceramic base material is stable even at temperatures far higher than those encountered under the most severe conditions of operation. In fact, it has been tested satisfactorily at temperatures of up to 2,000 deg. F. To give the necessary friction, long-wear and non-abrasive qualities other ingredients are incorporated in the linings.

Because Cerametallic brake linings will stand up to these high temperatures, brake loading may be increased without danger of lining disintegration. In fact, experience has shown that the effective brake capacity can be increased by as much as 50 per cent. Another advantage of this material is that no fade is experienced. With some of the more conventional materials, hard crust-like deposits are formed at high temperatures and weld to brake components. This causes erratic brake action. Hard deposits of this nature are not formed with the new material so that brake operation is much more predictable and grabbing is not experienced.

The surface of Cerametallic brake linings does not glaze and change its frictional characteristics when used. Moreover, these linings remain fully effective until they are worn down to the smallest usable thickness. The thermal conductivity of the material is higher than that of more conventional linings, so that, for a given friction loading, brakes lined with it run cooler than do the older units. This has an additional advantage in that the effectiveness of the shoes as a heat reservoir is greater than in brake units in which the rate of heat transference to the shoes is slower. Moreover, the lining material has a higher specific heat than resin based materials, so that for a given temperature, its capacity for storing heat is larger.

Less dust is formed with the Cerametallic material than with resin based linings. It is claimed that because the new lining has a less abrasive action, the metal surfaces which contact the linings may be expected to have up to 10 times the life that is experienced with normal linings. In addition, laboratory experience and operating tests have shown that Cerametallic lining wears more than five times longer than the best resin based material. Because of the slower rate of wear, brakes may be operated for longer periods without adjustment. It is also stated that no running in is required, and that the lining is fully effective on the first brake application. (2042)

New U.S.A. Patent Act

A REVISED patent act, known as Public Law 593, came into force in the United States of America at the beginning of this year. Patentees will naturally seek the guidance of patent agents in taking out American patents, but a few notes may be given on the more interesting points of the new law.

One of the striking changes is that under the new law, mistakes about joining, or not joining, co-inventors may be corrected, so long as they did not arise from deceptive intent. Under the old law, correction was impossible, and errors of this kind could invalidate a patent. Similarly, if a co-inventor refuses to sign the necessary patent documents, or cannot be found or reached after diligent efforts, the other inventor can sign for both himself and his co-patentee, providing the Commissioner of Patents is satisfied that such action is necessary.

Where, for one invention, patents have been applied for in countries other than the U.S.A., it will be necessary, so far as regards patents filed in the U.S.A. after 1st January, 1953, to stop issue of the foreign patent until the U.S. patent has been filed, when the U.S. filing takes place more than twelve months after the filing of the foreign patent. What is very important, especially for British applicants who rely upon British provisionals to support the basic priority dates, is that these provisionals should be full and clear and, where possible, they should be illustrated by drawings.

Applicants will now be put to the expense of filing priority documents, namely, certified copies of the original foreign application, specification and drawings. If not originally in English, these will have to be accompanied by a translation. The new law states definitely that a new use of a known process, machine, and manufacture composition of matter or material is patentable.

SIMMS SPE-6A INJECTION PUMP

A New Unit for High Speed, Six Cylinder Diesel Engines

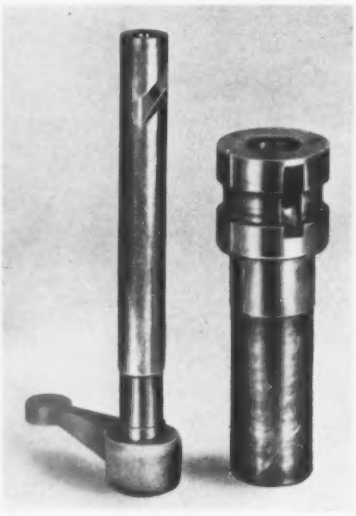
WITHIN a month or so, the Simms injector pump for six cylinder engines will be in full production at the East Finchley works. It incorporates several noteworthy features, including a pneumatic governor with a second air connection for damping at low speeds, a simplified spill control profile on the plungers which are operated by an improved control system, and a special cam profile to prevent running back. A solenoid operated cut-off and a fuel pressure actuated enrichment device for starting are optional extras. All these features are new, with the exception of the governor and spill control system which were first used on the four cylinder version, introduced about 9 months ago.

The unit described here is the SPE-6A 75 S334. This specification is derived as follows:

SPE	Simms pump, self-contained type
6	Number of elements
A	Size of pump (38 mm shaft centre height)
75	Plunger diameter in 1/10 mm
S334	Specification number relating to a particular engine application (in this case the Leyland Tiger Cub).

In general, the latest SPE range is intended to meet the requirements of high speed, direct injection engines of about 1·25 litres per cylinder. Of recent years the demand for this type of power unit has been increasing steadily, and it is now high enough for production to be undertaken in quantities sufficient to bring the cost per unit to an appreciably lower level than hitherto. These economies are not only attributable to greater efficiency in production, but also to the spreading of development costs over a larger number of units.

The fact that development of injection equipment represents a large proportion of the cost of this type of unit is not widely realized. Almost as much development work is necessary on the pump as on the engine and, because the cost of the pump is appreciably less than that of an engine, the development charges tend to be a greater proportion of the total. However, this tendency may be less marked over a long period of time when one type of injector is fitted to a number of different makes of power units, although additional development must be done for each new application.



A plunger and barrel

When the introduction of the new SPE range was first contemplated, it was decided after careful consideration to adopt the conventional multi-element in-line layout rather than the single element type with a rotary distributor. While realizing that opinions expressing preference for one or the other type could be regarded as controversial, Simms Motor Units consider that the single element design has yet to be proved adequate for the heavy duty imposed on it in direct injection applications. For not only does the single plunger have to supply all six cylinders, but also the pressures under which a direct injection unit works are about double those of the indirect units. Because of these higher pressures, nominally 175 atmospheres in the SPE-6A pump, the higher rate of wear that would almost inevitably have occurred had a single plunger been employed, would have been all the more critical.

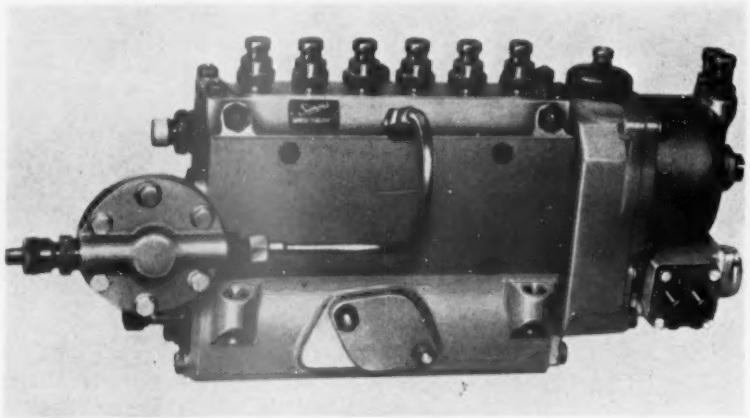
The pump unit

The combined pump and governor forms a simple, compact and robust unit. The overall length of the D.T.D. 424 body casting is 220 mm, and of the governor unit 123 mm, while the width and height of the body are 104 mm and 180 mm respectively. Except where metric threads are standard practice, unified threads are employed throughout.

In common with all the other Simms injection pumps, the body is split horizontally in such a manner that the six element assemblies comprising the plungers, barrels, springs and their retaining washers, and the delivery valves are carried in the upper portion. This is a good feature from the servicing point of view, since it improves accessibility. The lower part contains only the camshaft assembly, tappets and control rod. Five $\frac{1}{4}$ in diameter studs are mounted in the top face of the lower portion to hold the two parts together.

A 16 s.w.g. steel inspection cover is held down on its joint face on the side of the tappet chamber by eight $\frac{1}{4}$ in diameter bolts. The top part of the joint face is on the upper portion of the body and the remainder is on the lower portion. A Neo-K-Tex joint washer is employed to give a perfect seal and to allow for any slight misalignment of the two parts of the face.

No provision has been made for lubrication during service, since this function is performed by fuel oil leakage past the barrels and plungers. Oil filler plugs always become covered with dirt, and it was considered that the danger of foreign matter entering the unit when topping up with oil during



The Simms SPE-6A injection pump with a fuel pressure actuated enrichment device and solenoid operated cut-off fitted

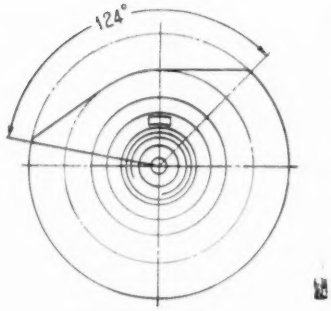
service, was best obviated in this way.

In any case, such servicing operations are not always easy on underfloor engines because of inaccessibility. However, when fitting a new or reconditioned pump, in addition to assembling of all components with oil, $\frac{1}{8}$ pint of engine lubricating oil is poured into the unit. A banjo union for an overflow pipe is screwed into a steel insert in a boss cast on the side of the cam box.

The case-hardened En 32 camshaft is unusually sturdy, being 30 mm diameter. At each end it is machined down to 20 mm diameter to carry the inner race of a ball bearing. The two bearing housings are spigoted into the ends of the cam box, and each is secured by four $\frac{1}{4}$ in diameter set bolts. Provision can be made to drive the unit from either end, and in some models the bearing housing at the end opposite to the drive is integral with the governor casing.

A $\frac{1}{2}$ in UNF thread is provided on the driving end to pull the coupling on to the 20 mm long, tapered portion of the shaft which is slotted for a Woodruff key. The taper is 1 in 5 on the diameter. An Angus, lip-type oil seal is retained in the bearing housing at the driving end by a pressed steel cup held in position by the bolts that secure the bearing housing over which the cup is fitted. Between the oil seal and the bearing, an oil baffle is clamped between a shoulder in the housing and the outer race. At the end opposite to the drive, the bearing housing forms the seal, since it is not drilled for a shaft extension. The camshaft end float is 0·002-0·006 in. It is adjusted by shims between the inner races and the shoulders on the camshaft.

The cam profiles are somewhat unusual. Continued running back, which hitherto has been troublesome on some engines, has been made impossible by extending the dwell angle to approximately 236 deg. Thus, should the



The new cam profile

engine run back, injection occurs at the bottom of the exhaust stroke. The base circle diameter is large, 36 mm, to give the high rate of injection necessary for direct injection engines.

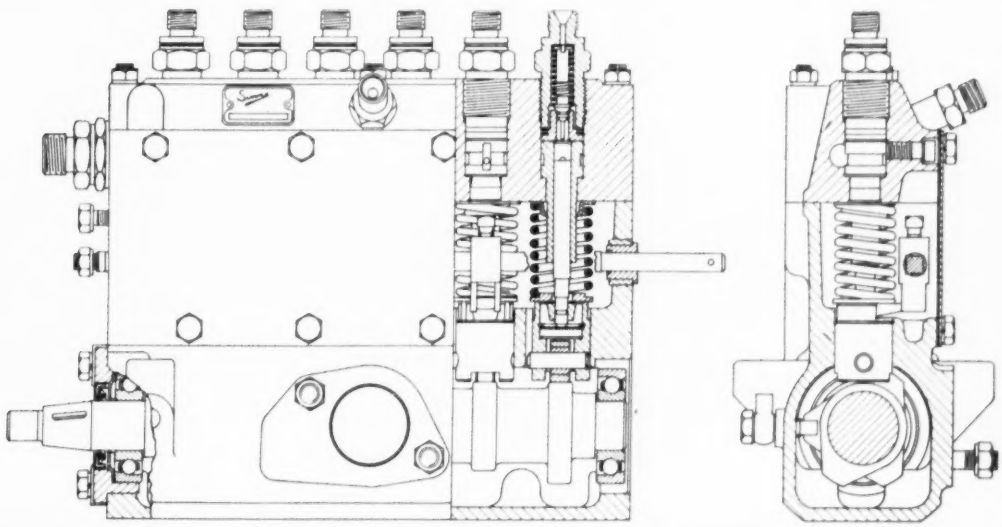
Roller-type tappets of case-hardened En 32 are employed. Their barrels are 24 mm diameter, and they are carried in guides 23 mm long. At the upper end, a slot is machined on one side to clear the lever operating the spill device. This end is also counterbored to carry the discs for phase adjustment. The discs are of case-hardened mild steel, and are held in position by Seeger circlips. They are supplied in three thicknesses: 4·6 mm, 4·7 mm and 4·8 mm, each 0·1 mm interval represents $\frac{1}{4}$ deg camshaft rotation. Number 1 tappet is treated as the master and the remainder are phased to it.

The rollers are 18 mm diameter by 9 mm long. For both the rollers and the pins, En 31 was originally specified, but because of the current material supply position, En 351 ($\frac{1}{4}$ per cent nickel chrome case-hardening steel) is used. Floating bushes of through-hardened, KE 637 high carbon manganese steel are carried in the rollers. The pins are 7 mm diameter. They float in the tappets in such a manner that one end projects about 3 mm into a vertical groove in the tappet housing to provide location against rotation.

The injector elements incorporate several novel features. One of these is the spill control. This control is effected by means of an En 8 stamped lever pressed on to the shouldered lower end of each through-hardened En 31 plunger. The other end of this lever engages in a forked block on a square section control rod in such a manner that its vertical movement is unrestricted. The extreme ends of the rod are of round section where it is carried in the end walls of the tappet chamber. The end adjacent to the governor control unit bears in a flanged brass bush, while the other bears directly in the aluminium housing of the enrichment stop, which is spigoted into the end wall. Longitudinal movement of the control rod causes angular motion of the arm and partial rotation of the plunger. Calibration of the fuel delivery to each element is effected by slackening a clamping screw that holds the forked block on the control rod. The block may then be slid along the rod, and the screw re-tightened.

This method of control is an improvement on the more conventional mechanism in which there are two clearances, one at the rack and pinion and the other at the control sleeve. In the new device there is only one clearance and this is at a greater radial distance from the plunger axis, so the accuracy and sensitivity of calibration is improved. Another advantage of this arrangement is that friction which might result in sluggish control movement is also reduced.

The spill control edge at the head of the plunger is another noteworthy feature of the injector element. Instead of the more usual rather complex curved spill edge, a straight edge is incorporated. This edge is formed by machining a straight groove across the periphery of the plunger at an angle of approximately 40 deg to its axis. A



In this illustration of the new pump, the earlier cam profiles are shown

radial hole from the bottom of this groove communicates with an axial one drilled from the upper end of the plunger. Not only is the straight groove less costly from the production point of view but, since only a relatively small portion of the peripheral surface of the plunger is cut away, the bearing area at this end is larger, so that the rate of wear and possible paths for leakage are reduced. The upper end of the plunger spring bears against a pressed steel washer under the upper part of the body and the lower end against a shouldered washer above the boss of the spill control lever. It is shot peened and then zinc electroplated to prevent corrosion.

Selective assembly ensures that the best possible fit is obtained between the plunger and the En 31 barrel. The top end of the barrel is ground to about 17 mm outside diameter for a length of about 15 mm. Then it is reduced to 14 mm diameter and ground for a length of 9 mm and the remainder is rough turned down to about 13 mm diameter. The ground peripheries are a push fit in the housing. Immediately below the large diameter portion, the periphery of the barrel is relieved so as to reduce wear on the corner of the grinding wheel used in production.

The large diameter top end forms a deep flange which seats on a shoulder in the upper portion of the body, and it has an annular groove around it from which two radial holes spaced 180 deg apart communicate with the inner bore of the barrel to form the main and auxiliary ports. A vertical groove is milled across the annular one so that its vertical axis is in the same plane as the axis of the auxiliary port. The dowel end of a cheese-headed set screw, carried horizontally in the upper portion of the body, engages in this groove to furnish location against rotation. The head of this screw is countersunk behind the tappet chamber cover to keep it clean and to prevent it from becoming loose and falling out. On the opposite side of the barrel, the fuel gallery breaks into the barrel housing at the level of the annular groove. Not only does this arrangement reduce the overall width of the unit, but it also obviates the necessity for drilling transverse passages to pass the fuel to the injector element.

A conventional check valve assembly is screwed into the top of the body to hold down the barrel. It consists of an En 33 valve and guide and the valve holder. The lower end of the guide is flanged to seat on top of the barrel. The valve holder is screwed down on to this flange, and the seal is made with a Tufnol washer. A coil spring seats on a shoulder round the top of the valve and is housed in the body, together with a cylindrical volume reducer inside the spring. The lower end of the valve is fluted where it engages in the guide.

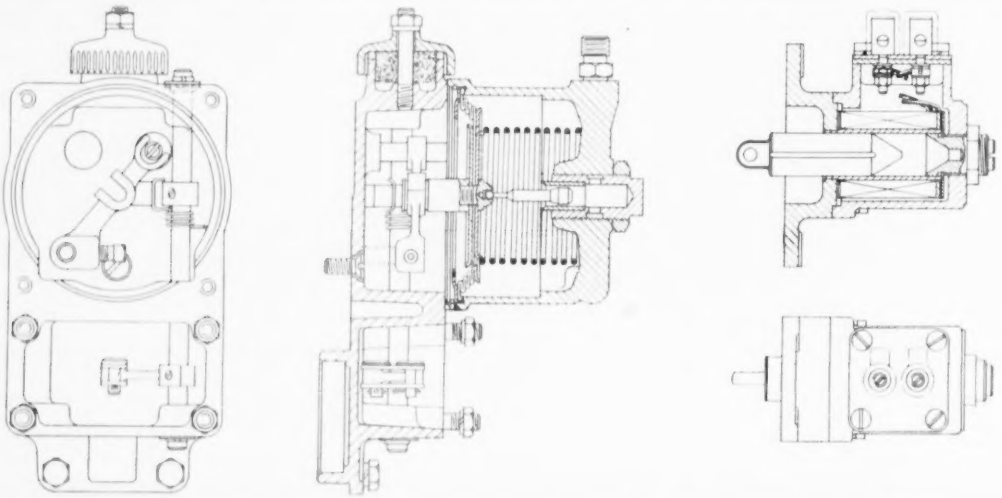
The governor and cut-off

There are three main parts of the pneumatic governor. One is the throttle unit which in most, but not all, applications is made by the engine manufacturer. The second is the housing for the control linkage; it is mounted on the end of the pump unit. Bolted to this housing is the third part, the diaphragm chamber.

The housing for the control linkage is an aluminium die-casting. In the form shown in the illustration, it incorporates at its lower end the camshaft bearing housing and a boss on which is mounted a solenoid cut-off control. A manual control may be fitted as an alternative, but solenoid operation in some installations may avoid long control links that may work under unsatisfactory conditions because of frame distortion. The solenoid control is also most convenient for stationary engines required to be stopped automatically at a predetermined time. On top of the linkage casting is a breather aperture containing steel wool to prevent the entry of foreign matter. A plastics cap is held down over this aperture by a centrally positioned stud and nut.

A cylindrical chamber, the axis of which is horizontal and in line with that of the diaphragm housing, is formed in the top portion of the casting. In the chamber is the link between the spill control rod and the diaphragm mechanism. This link is at about 50 deg from the horizontal in the transverse vertical plane. On one end is a boss carrying the spill control rod. A radially positioned set screw in the boss clamps the link to the rod. The other end is forked and slides freely on a guide pin screwed into a boss on the wall of the chamber. At the centre of the link is another boss in which is a U-shaped slot cut in such a manner that it can be engaged in a groove round the end of a rod attached to the centre of the diaphragm.

Offset about 1⅛ in to one side of the axis of the chamber is a vertically positioned spindle. Its upper and lower ends are each carried in a boss in the casting. Immediately above the top boss and below the bottom one are split pins, each passing through a hole in the spindle to retain it in position. This spindle carries two levers, each clamped to it by a set screw radially positioned in its boss. The end of the lower lever is pinned to the operating rod of the solenoid cut-off control, while the upper one bears against the end of the rod carried on the centre of the diaphragm. Thus, when the solenoid control is operated, its motion is transmitted through these two levers and their spindle to the end of the diaphragm rod, and thence through the transverse link to the spill control rod. This rod, with its operating forks, moves the lever at the lower end of each plunger which is rotated to a position such that as the higher of the two spill ports is on the point of being closed by the top of the plunger, the lower one is opened by the uppermost tip of the spill groove.

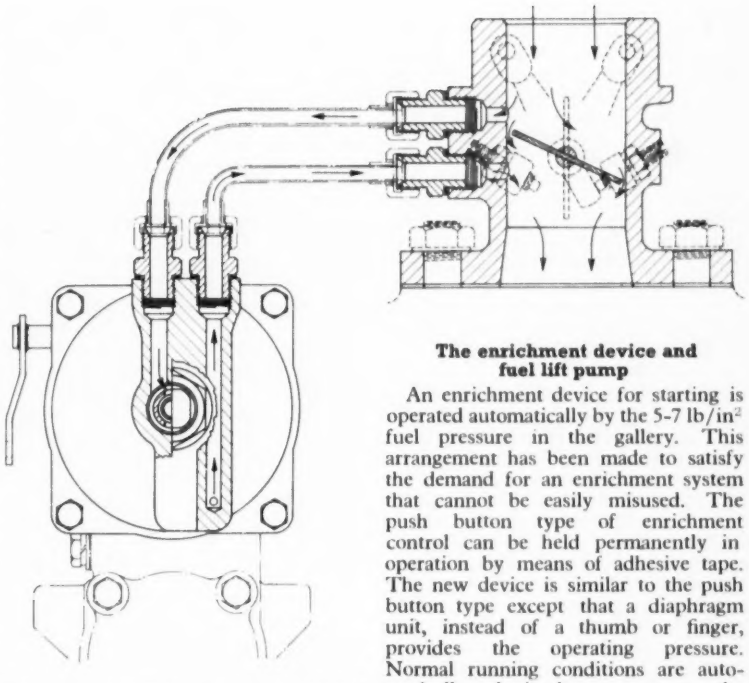


From left to right are: the governor linkage, another view of the linkage together with the diaphragm chamber, and two views of the solenoid cut-off control

The diaphragm chamber is a cup-shaped aluminium die-casting. Its open end is flanged and bolted to the linkage chamber. A synthetic rubber bellows-type diaphragm is employed and, before assembly, it is held in position between a shoulder round the inner periphery of the chamber and a pressed steel ring, by a wire circlip. When the diaphragm chamber is bolted home, a spigot on the linkage chamber bears on the pressed steel ring and clamps the diaphragm firmly against the shoulder. The centre of the diaphragm is clamped between two dished steel plates, and the whole assembly is mounted on the shouldered end of the central operating rod. The other end of this rod, as has already been stated, engages the slot at the centre of the transverse link. The governor spring is positioned between the closed end of the chamber and the dished plate at the middle portion of the diaphragm. Projections are pressed out from the plate to locate the spring symmetrically about its centre. Bosses in the casting locate the other end of the spring.

A noteworthy feature of the governor is the damping device to prevent hunting at low engine speeds. The damping valve assembly is in a cylindrical housing positioned co-axially with the diaphragm chamber in the centre of the closed end. This housing is screwed into position and secured by a lock nut at its outer end so that adjustment can be made easily. An accurately ground valve guide tube is pressed into its inner end. Round the outer periphery of the housing, approximately mid-way between its ends, is an annular groove from which two holes are drilled radially to communicate with the interior. The valve itself is a steel rod, one extremity of which is ball-ended and is carried in an axial drilling in the diaphragm rod, in which it is held by peening. On the other end of the valve is an integral tapered head.

From the illustration it can be seen that when the engine is slow running and the pressure in the diaphragm chamber is low, the valve head is in line with the axes of the radial holes in the guide assembly. Further lowering of the pressure, tending to extend the diaphragm still more, brings the reduced diameter of the tapered portion of the head opposite to these holes. This relieves slightly the pressure in the diaphragm chamber since, with the throttle in the slow running position, the holes are in communication with the atmospheric pressure side of the throttle unit. The damping valve, therefore, acts as a buffer and prevents undue oscillations of the diaphragm at low speeds.



Two pipe connections are incorporated between the governor and the throttle unit

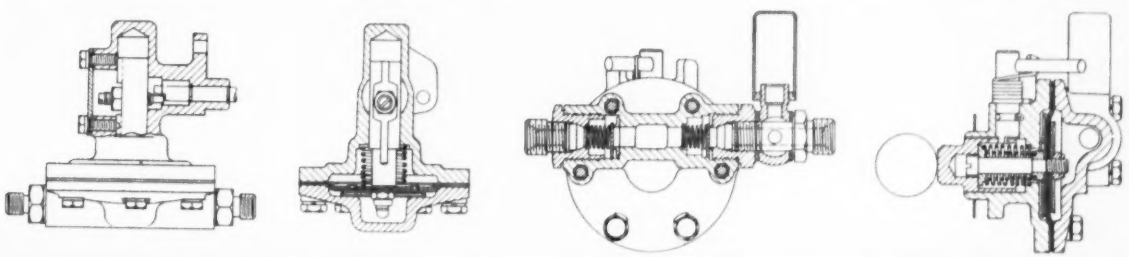
The action of the throttle unit is simple, though ingenious. Two pipe connections are employed, one from a union communicating with a drilling to the annular groove round the damper valve housing, and the other through a similar union and drilled passage to the diaphragm chamber. The pipe from the diaphragm chamber is connected in the usual way to the low pressure side of the throttle butterfly. The damper pipe, however, communicates with a hole drilled in such a position that with the throttle nearly closed it is on the atmospheric side of the butterfly, but as the throttle is opened, the edge of the butterfly disc swings past it so that low pressure is communicated through the damper valve. Thus, the damping system does not interfere with the normal operation of the governor.

The enrichment device and fuel lift pump

An enrichment device for starting is operated automatically by the 5-7 lb/in^2 fuel pressure in the gallery. This arrangement has been made to satisfy the demand for an enrichment system that cannot be easily misused. The push button type of enrichment control can be held permanently in operation by means of adhesive tape. The new device is similar to the push button type except that a diaphragm unit, instead of a thumb or finger, provides the operating pressure. Normal running conditions are automatically obtained as soon as the engine starts.

The diaphragm chamber is similar in shape to that used on the fuel lift pump, the diaphragm being clamped between two circular die-castings. One of these castings has a fuel inlet and outlet incorporated, while the other houses the plunger and return springs. The return spring bears at one end against one of two dished steel plates, between which the diaphragm is clamped, and at the other end against the casting. The operating plunger is co-axial with the spring and is threaded and shouldered at one end where it is passed through the centre of the two plates and the diaphragm which are pulled against the shoulder by a self-locking nut. About $\frac{3}{4}$ in. from the other end, a dowel ended set screw is passed diametrically through the plunger and locked in position by a lock nut which seats on a flat that is machined on the plunger. The dowel end projects approximately $\frac{5}{16}$ in beyond the other side of the plunger to form the enrichment stop.

In the illustration, this stop is shown in the rich position for starting, the diaphragm and plunger being withdrawn under the influence of the



On the left two views are shown of the fuel operated enrichment device, and the two views on the right are of the fuel lift pump

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return spring so that the stop engages in the step machined from the end of the spill control rod. When the engine starts, the diaphragm under the influence of the fuel pressure, pushes the plunger into such a position that the stop is in line with the end of the spill control rod.

The fuel line to the inlet on the diaphragm chamber cover is taken from the gallery of the injector pump. An outlet also is incorporated in the cover, and in it is a restricting orifice to bleed a small quantity of fuel back to the tank. This arrangement provides a constant bleed for the system, so that no bleed screws have been incorporated in the pump unit. From the servicing point of view, this feature is particularly attractive in underfloor engined vehicles, in some of which accessibility is not all that might be desired. By altering the size of the bleed orifice, the timing of the trip-out of the enrichment device may be varied.

The fuel lift pump is fairly conventional, but it incorporates one or two interesting features. The diaphragm is not actuated directly by the eccentric, but the motion of the tappet is transmitted to two co-axial coil-springs. The cylindrical body of the tappet is drilled out and shouldered internally, so that it surrounds the coil springs and the shoulder bears against the spring retaining washer. A bolt is passed through the centre of the washer to hold together the spring and diaphragm assembly.

Both springs bear at one end against this washer, but the outer spring, which maintains the tappet in contact with the eccentric, bears at its other end against a shoulder in the pump body. The inner spring, on the other hand, bears against another washer clamped between a shoulder on the central bolt and the diaphragm, the whole assembly being secured by a self-locking nut on the end of this bolt. When the outer spring is compressed by the action of the tappet and eccentric, the spring retaining washer in the tappet is free to slide along the shank of the bolt against the action of the inner spring. Thus, should the fuel pressure on the outer side of the diaphragm become excessive the inner spring, through which the motion is transmitted from the tappet to the diaphragm, will be compressed so that the diaphragm cannot be overloaded and its life is increased.

In the inlet and outlet, Tufnol discs are employed in the non-return valves, and they are held down on their seats by small compression springs. A banjo union on the outlet is connected to an air vessel to damp out pulsations. The internal dimensions of the air vessel are approximately ¼ in diameter and 1½ in long. A lever type hand operating device is incorporated for starting after a long period of standing, and it is held out of engagement when not in use by a spring coiled around its spindle. A projection on the inner end of the spindle engages in a slot in the tappet to actuate it in the usual manner.

ROLLING BEARINGS

Behaviour at Very High Running Speeds

AN article, by G. Getzlaff, published in the November 1952 issue of the *Engineers' Digest*, gives some results of tests carried out on rolling bearings. The types used were Series 62 light-load radial deep-groove ball bearings, for shafts of 10-35 mm diameter, and Series NL35 radial cylindrical type light-load roller bearings for shafts of 35 mm diameter running at speeds varying between 20,000 and 70,000 r.p.m. One test rig was driven by means of an A.C. motor used in conjunction with a change-gear unit, the second by a compressed air, open jet type turbine with infinitely variable speed up to about 100,000 r.p.m. The loading conditions of high speed bearings differ from those of normal ones in that the bearing load due to external forces is smaller since at high speeds the rated power is obtained with lower torque, and the internal loads due to centrifugal forces on the cage and rolling bodies, and those due to sliding friction at high speeds, become more important.

Among the operating factors affecting high-speed bearings, temperature stability is taken as the criterion. An increased supply of lubricant decreases the bearing temperature, although a temperature rise does not invariably follow from a reduced oil supply. With a constant flow of lubricant, the temperature increases linearly with running speed. Additional axial load raises temperature in deep-groove ball bearings, while a small unidirectional load of this type promotes more stable running conditions by damping out impacts and vibrations. Radial clearance considerably affects the running temperature; a change in clearance from 20 to 30 microns with NL35 roller bearings causes a temperature variation of over 35 deg C.

Lubricant viscosity is important, as bearing temperature increases roughly as the square root of the viscosity value in poises. Greases were found satisfactory at lower running speeds, and silicones, with their nearly constant viscosity, might also be suitable. The oil mist system, using a small vane-wheel was found to give the best all-round results. Aluminium alloy proved superior to brass or specially treated wood for cage materials. Smooth running conditions and bearing life depend on correct fitting tolerances, and tests indicate the following minimum values for radial fitting clearance:—

Bore	10 mm	20 mm	35 mm	35 mm
Clearance	15 μ	15 μ	30 μ	30 μ

Operating temperatures depend on the thermal behaviour of the bearing materials and the lubricants used. The temperature measured at the outer ring should not exceed 130 deg C. (*M.I.R.A. Abstract No.* 6134.)

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

The following meetings will be held during April:—

LONDON

Tuesday, 14th April, 5.30 p.m. General Meeting at Storey's Gate, St. James's Park, S.W.1. Paper: "The Jaguar Engine," by W. M. Heynes, M.I.Mech.E. (Member of A.D. Council).

NORTH-EASTERN CENTRE

Wednesday, 15th April, 7.30 p.m. General Meeting in the Chemistry Lecture Theatre, The University, Leeds. Paper: "Research and the Engineering Process, with Particular Reference to the Automobile Industry," by H. E. Merritt, M.B.E., D.Sc.(Eng.)(Lond.), M.I.Mech.E. (Member of Council).

NORTH-WESTERN CENTRE

Monday 27th April, 7.15 p.m. General Meeting at Leyland Motors Limited, Leyland, Lancs. Paper: "Some Problems Arising from the Wider Use of Small Diesel Engines," by J. H. Pitchford, M.A. (Cantab.), M.I.Mech.E.

WESTERN CENTRE

Thursday, 9th April, 6.45 p.m. General Meeting at Fortts Restaurant, Milsom Street, Bath. Paper: "Brake Linings and their Performance," by J. G. Remington, A.M.I.Mech.E.

SMALL CAR PRODUCTION

The Manufacture of Engines and Transmissions for the Austin 'A30'

Part II

THE methods employed at present by The Austin Motor Co. Ltd. in the manufacture of the gears for the transmissions of their "A30" vehicles do not call for detailed comment. They follow conventional practice for the production of high quality gears on machines of standard types that may be found in every British automobile factory. For many of the transmission details other than the gears, however, the Company employ specially developed machines and methods that show a definite break from conventional practice. Without any doubt, the most interesting feature of the production planning is the wide use made of special purpose, in-line and rotary transfer machines that incorporate Austin unit heads.

Gearbox machining

An integral aluminium die-casting is used for the gearbox and bell housing. Practically all the machining is carried out on machines equipped with Austin unit heads, and although it is a fairly complex component to machine, the amount of handling required has been reduced to a remarkable degree through the incorporation of two automatic in-line transfer machines in the machining line.

The first operation, machining the flange face, is carried out on a machine with an Austin head from which the spindle drive mechanism has been omitted and which has traverse motion only. A special pot chuck is mounted on the rotating table of the machine. The casting is mounted in this chuck with location taken from bolt bosses on the rear face of the flange and is centralized from the bore. Fig. 23 shows the set-up on this machine, which has a fully automatic cycle.

After the flange has been faced, the casting is transferred to an Austin horizontal unit head machine with a multi-spindle drilling attachment. There are 11 spindles, nine carrying drills and two carrying a combination drill and reamer. This machine drills 11 holes in the flange face and reams two of the holes to act as locations for subsequent operations.

From the horizontal multi-spindle drilling machine, the component is transferred to a 13-station in-line automatic transfer machine. This machine, in common with the other in-line transfer machines to be discussed in these notes, is equipped with Austin unit heads, but the base and the transfer mechanism are designed and manufactured by James Archdale & Co. Ltd., Blackpole, Worcester. The work fixture is mounted on a standard platen that is used in conjunction with all the in-line transfer machines. Loading is effected at the first station with location taken from the machined face of the flange and from the two reamed holes in the flange. The component is loaded into the fixture with its longitudinal axis at right angles to the longitudinal axis of the machine and with the flange to the right.

On this machine, and on other in-line transfer machines, one of the problems has been to arrange guide bush plates that will give adequate support to the drills or reamers and yet also be clear of the casting when indexing is being effected. Various methods have been adopted to bring the guide bush plates into close proximity to the work for the actual machining operation and then, when necessary, to retract the bush plates clear of the work for indexing. Where the guide bush plates are not fixed, their movements are effected automatically in synchronization with the remainder of the machining cycle. Here, it may be pointed out that the coolant system is also interconnected with the cycle of each individual station in such a manner that as soon as the tools stop cutting, the coolant is automatically cut off. The supply is automatically started again immediately before the tools start cutting in the next machine cycle.

Fig. 23. Austin unit-head machine for facing the bell housing flange of the gearbox casing

At the first working station, see Fig. 24, on this 13-station machine, there is a single spindle boring unit at the left of the transfer line. It is tooled for boring the main shaft holes in the rear wall of the gear casing and in the dividing wall between the gear casing and the bell housing. The right-hand station is vacant. At the second working station there is a horizontal 11-spindle drill head at the left. It drills two rod holes, seven tapping holes, a reverse shaft hole and a layshaft hole in the rear face of the casing. Also at this station there is a vertical drill unit with eight spindles for drilling tapping holes in the side cover face. Fixed guide bush plates are used at this station.

Two horizontal multi-spindle drill heads, one at the left and one at the right of the transfer line, are used at the third working station. The left-hand head is tooled for drilling five holes, including a rod hole and a reverse shaft hole in an inside boss. For drilling the reverse shaft hole, a special retractable guide bush plate is used. It is shown in Fig. 25. This guide bush plate is attached to the piston of a hydraulic cylinder. At the appropriate instant in the machine cycle, pressure is admitted to the top of the cylinder and the guide bush plate is moved down to bring the bush into position to support the drill. After the drill has been retracted, the bush plate is automatically returned to the up position to allow the component to be indexed to the next station. The right-hand head at this station drills a rod hole, three oil holes and two tapping holes in the front face of the component.

Two blow-out and gauging units are mounted at the fourth working station. These units not only clear any swarf from the tapping holes but also gauge the holes for depth. If for any reason any hole is not clear to the full depth, the gauging unit is so interconnected with the machine mechanism

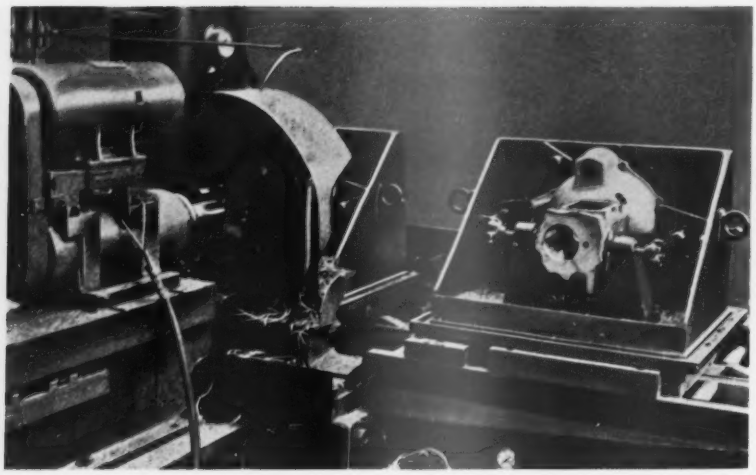

Fig. 24. The loading end of a 13-station in-line transfer machine for the gearbox casing

that the machine is stopped. This, of course, prevents the breakage of taps and the production of faulty work. A horizontal unit is used at the left to blow-out and gauge nine holes in the rear face, while a vertical unit at the right blows-out and gauges eight holes in the side cover face.

The holes blown-out and gauged at the fourth working station are tapped at the fifth station from a horizontal head at the left and a vertical head at the right. Multi-spindle drill heads are used at both sides of the next station. That at the left is used to drill three tapping holes in the starter housing, while the right-hand head drills nine holes in the front face of the component, including two rod holes and a layshaft hole. A spring-loaded guide bush plate is mounted on the right-hand head.

Two horizontal unit heads are mounted at the seventh working station. The head at the left of the transfer mechanism is tooled for tapping the three holes in the starter housing while the right-hand head has a blow-out and gauge unit for clearing and checking the tapping holes in the front face of the casting. These holes are tapped from a horizontal unit head at the next station. The left-hand head at this station is tooled for opening out three rod holes, a reverse hole and the layshaft holes that were drilled at previous operations. A hydraulically-operated retractable guide bush plate of the type described earlier is used in conjunction with this head.

At the ninth working station a compound slide unit at the left is used for milling the layshaft bosses. The right-hand station is vacant. Only a left-hand head is used at the tenth working station. It is adapted to act as a rise-and-fall miller for machining the rear face to length. The final working station on this machine, see Fig. 26, is used for milling the side cover face in the specified relationship to the main bores. A bridge frame, with the milling head mounted on it, is built across the transfer line at this station.

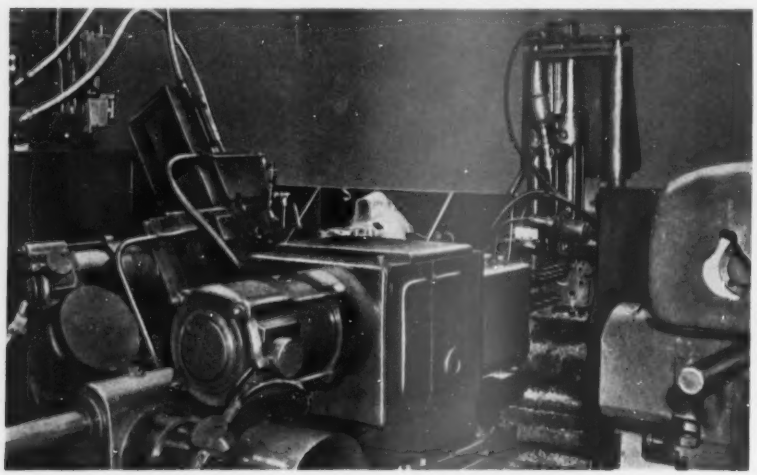

Fig. 25. Hydraulically operated retractable guide bush plate on the machine shown in Fig. 24

At this stage the component is transferred to a standard milling machine on which the front face is milled in relationship to the rear face. It is then passed to a two-way horizontal unit boring machine which is tooled to counterbore the rear main bearing and slightly countersink this bore from one head while similar operations are carried out on the front main bearing from the other head.

The component then goes to the second in-line automatic transfer machine, which has 13 stations. It is shown in Figs. 27 and 28. Loading is effected at the first station with the shaft axis of the casting in line with the longitudinal axis of the machine and with the rear end leading. Location is taken from the machined rear face and from the two location holes in the front flange.

The first working station on this machine, see Fig. 28, has a horizontal unit head to the left of the transfer line. It carries a milling attachment and a special form drill for milling the clutch shaft boss in relation to the main bores and for drilling a hole. There are two horizontal unit heads at the right. They are disposed at an angle to the horizontal, one to drill down and the other up. Each head carries a $\frac{3}{4}$ in diameter end mill for starting an angular hole. The drill guide bush plate for these heads is carried on two vertical columns. It is pneumatically operated to bring it into the working position before the tools advance and to withdraw it after the tools are retracted. This is necessary to allow the component to be indexed to the next station.

Only one head is used at the second working station. It is a vertical unit head with two spindles for drilling two holes at an angle of 30 deg to the vertical in the bottom of the box. As these holes are of considerable depth, a reciprocating unit head is used. The third working station has a two spindle horizontal unit head at the left for drilling two tapping holes in the clutch boss. At the right there are two horizontal unit heads disposed at the same angles as the right-hand heads at the first working station. They are tooled for drilling holes in the side cover face which were started at the first working station, and the guide bush plate arrangement is the same as at the first station. Because of the depth of the holes, reciprocating heads are used.

Only one head is used at the fourth working station. It is a two-spindle head at the left of the transfer line and is arranged to drill at an angle to the vertical. Two combined drill and pin facing tools are used to pin face and counterbore the holes drilled at the second station. The left-hand side of the fifth working station is occupied by a blow-out and gauging unit for the two tapping holes in the clutch boss, while from the right-hand side the holes drilled from the previous

angular heads are reamed to size. The guide bush plate arrangement for this is the same as at the first and third working stations.

There is only a single head at the next station. It is a two-spindle head tooled for reaming the holes drilled at the second station. At the seventh working station a horizontal unit head at the left of the transfer line is used for tapping two holes in the clutch boss, while at the right there is a two-spindle vertical unit head for drilling one hole in the bottom of the box and drilling and counterboring a hole in the reverse shaft boss. A retractable guide bush plate is used in conjunction with the right-hand head.

At the eighth working station a two-spindle tapping head at the left is used to tap two holes in the bottom of the box at 30 deg to the vertical, while at the right a single-spindle horizontal unit with a reciprocating head is used for drilling a hole in the side cover face. A spring-loaded bush guide plate is mounted on the reciprocating head. A two-spindle tapper at the left of the next station taps a drain hole and a hole in the reverse shaft boss, while at the right a horizontal unit head with reciprocating motion reams a hole in the side cover face. At the penultimate and final working stations on this machine, the left-hand stations are vacant. At each of these stations there is a single-spindle horizontal reciprocating drill head at the right of the transfer line, one to drill and the other to ream a hole in the side cover face.

After leaving the second transfer machine, the component is passed to a machine tooled for cutting away the rear layshaft boss and for skimming out a recess for the reverse wheel clearance. Thence the gearbox is passed to an Austin two-way horizontal multi-reaming machine. From a five-spindle multi-head at the right, three rod holes are reamed in the front and rear faces, the layshaft holes in the front and rear faces and the reverse shaft hole are reamed to size. There is a two-spindle head at the left. It is used to ream one dowel hole and combined drill and ream another. The machining of the gearbox casing and bell housing is completed on a vertical unit machine tooled for finish facing the flange in relation to the inside front face.

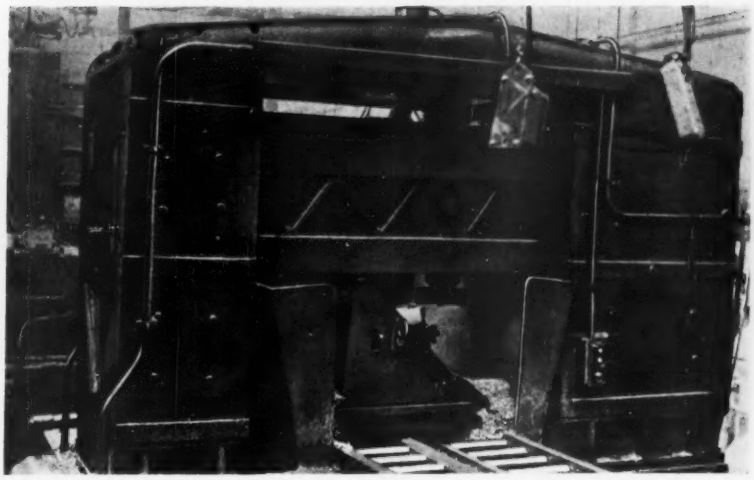

Fig. 26. The final working station on the 13-station gearbox casing transfer machine

Gearbox rear cover

An aluminium die-casting is used for the gearbox rear cover. The first machining operations are carried out on a Herbert No. 3A Auto lathe with a special pot chuck in which the component is loaded with the flange end out. From the first turret station on this machine two diameters are rough bored in the flange end and the bottom of the larger bore is faced, while at the same time the flange face is rough machined from the rear tool post. Following this, the two bores

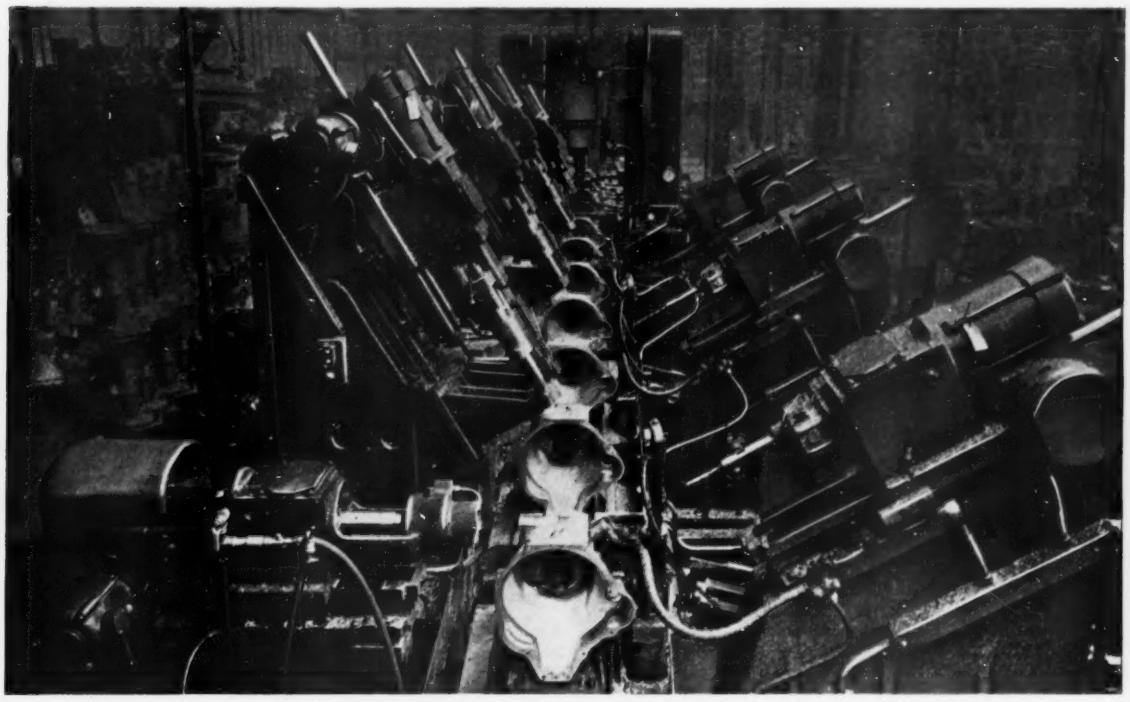

Fig. 27. The second in-line transfer machine for the gearbox casing

Fig. 28. The first working station on the machine shown in Fig. 27

are finish machined from the second turret face, while the flange face is simultaneously semi-finished from the front cross slide to leave an allowance for final truing.

Another Herbert 3A Auto lathe is used for the next operation. For this operation the component is loaded with the machined flange towards the headstock to allow machining to be carried out on the rear end. On this machine a combination tool-holder carries a boring bar with one boring tool and two knee tool-holders, each carrying a turning tool. These tools rough bore the rear end of the casting and rough turn two diameters. A radius and facing tool on the front cross slide faces the end and forms a radius.

A combination tool holder is used on the second turret slide. It carries a boring bar with three boring tools and two knee tool-holders each carrying a turning tool. The boring tools are arranged to semi-finish the bore roughed by the tool at the first station; to finish bore a recess; and to form a chamfer at the junction of the recess and the bore. The turning tools finish turn the diameters that were rough turned from the first turret face. Grooving tools on the rear cross slide then form two grooves in the turned diameters. Two knee tool-holders, each with a turning tool, are mounted on the third turret face. One tool turns a diameter and forms a radius while the other forms a chamfer on a diameter turned from the second turret face. Finally, a hole in an internal web is bored from the fourth turret face.

Before any further machining is carried out, the bore in the rear end of the casting is hand reamed and the bush is then pressed in. The component is then transferred to an Austin unit machine with a two-spindle vertical tapping and reaming head and a single spindle vertical fine boring unit. This machine has a rotary indexing table with four stations, one for loading and unloading and three working.

Location in the work fixture is taken from a bore in the flange end and from the flange face. Air-operated clamps are used, and the table is rotated by hand for indexing. At the first working station one of the spindles in the two-spindle head carries a special tap for cutting the oil return groove in the hole previously bored in the internal web. The component is then indexed and this hole is reamed from the second spindle of the two-spindle head. To complete the operations on this machine, the bush is fine bored to size from the fine boring unit head at the third working station.

Further machining is then carried out on a B.S.A. high-speed facing lathe. The component is mounted on a special mandrel that gives location from two machined bores and is tooled for facing the flange from the rear tool slide and facing to depth a recess in the flange end. This ensures that the elements machined on this lathe are truly at right angles to the axis through the bores.

The component is then passed to a multi-station in-line automatic transfer machine. Generally transfer machines are used to carry out machining some-

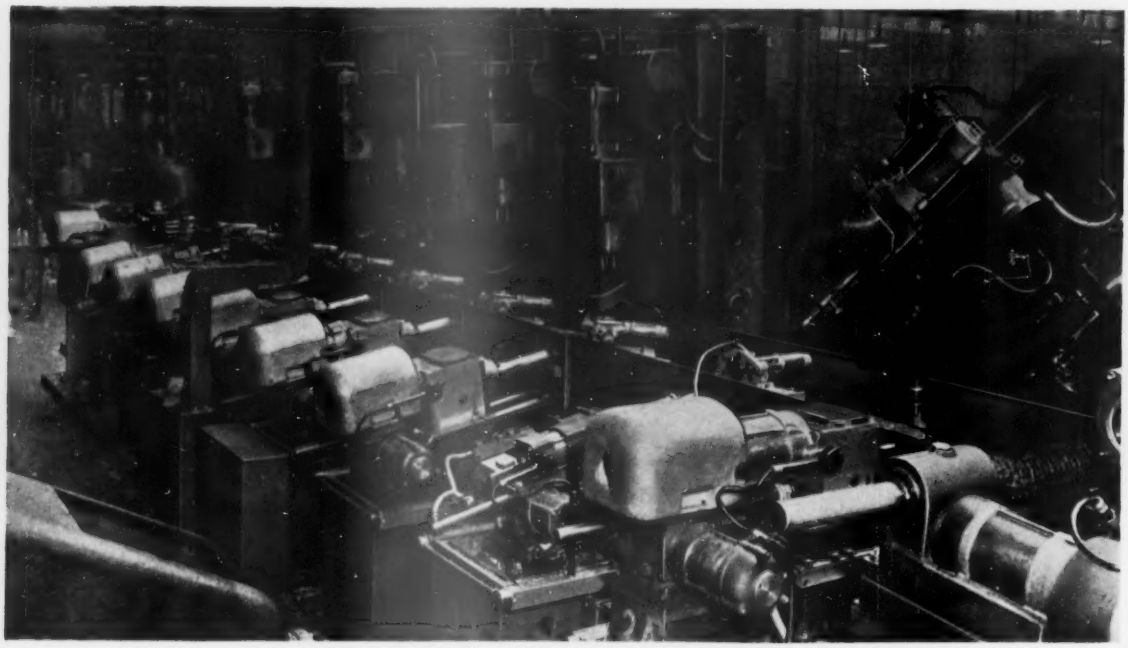

Fig. 29. Automatic in-line transfer machine for the gearbox rear end cover

where in the plane that is at right-angles to the transfer line, but for machining this component it has been necessary to arrange several heads to work at compound angles. There is also a second complication that is unusual on transfer machines. The casting is loaded into the work fixture with its longitudinal axis in line with the longitudinal axis of the machine, but some of the holes to be drilled are at such an acute angle in relation to the axis of the casting that it is impossible to set the heads for drilling to be carried out with the axis of the work coincident with the axis of the machine. For this reason provision is made for indexing the work during its passage through the machine. To effect this a special work fixture is used. The actual work-holding fixture is mounted on a circular indexing table which is mounted direct on the platen. At the appropriate points in the travel through the machine circular indexing is effected by means of cams. A general view of the machine is shown in Fig. 29.

At the left-hand side of the first working station there is a special horizontal unit head with a vertical milling head attachment for milling a face in relation to the centre-line of the component, while at the right there is a single-spindle drill unit arranged to drill a hole at 3 deg to the horizontal. A single-spindle head at the left of the second station carries an end mill for machining an elongated boss. The right-hand head at this station comprises a vertical unit head and a single-spindle drill head mounted on a special column to give a compound angular setting, see Fig. 30.

The third station has a single-spindle horizontal drill at the left for drilling a hole in an internal boss for the speedometer drive, while at the right a single-spindle tapper, mounted in the same manner and at the same angle as the right-hand head at the previous station, taps the hole drilled at the previous station. Between this station and the next working station the component is automatically indexed to bring the speedometer boss to the left-hand bank for drilling.

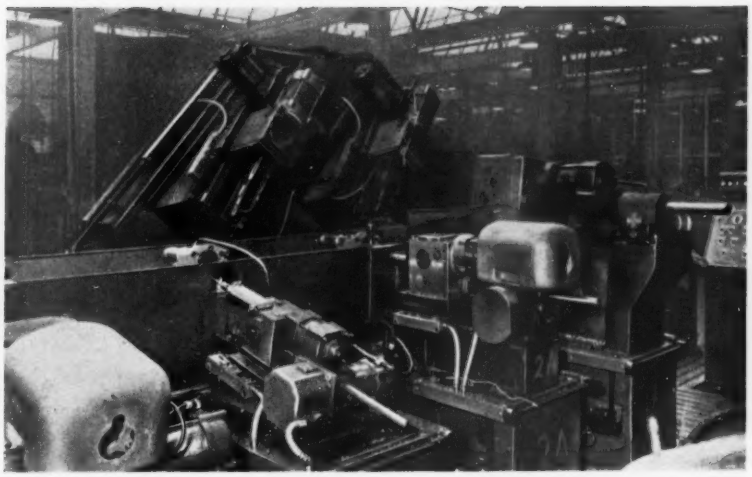

Fig. 30. The first working station on the machine shown in Fig. 29

At the fourth station the left-hand head carries a facing cutter for rough facing the speedometer boss, while a three-spindle vertical head at the right drills three tapping holes at a compound angle. These holes are countersunk at the next station, where at the left-hand side there is a single-spindle vertical drill head with a combination drill for drilling one hole and dimple spotting for another hole.

A blow-out and gauging unit is mounted at the right-hand side at the sixth station for clearing the swarf from, and checking the depth of three tapping holes, while at the left-hand side of this station there is a horizontal unit head with a combination tool-holder carrying a boring bar and tool for rough boring a diameter and a drill for drilling a hole that is later reamed. At the left-hand side of the seventh station there is a horizontal unit head with a single-spindle drill head carrying a facing bar, with two facing tools and a chamfering tool for machining the speedometer faces. Three holes are tapped from a vertical unit at the right-hand side of this station.

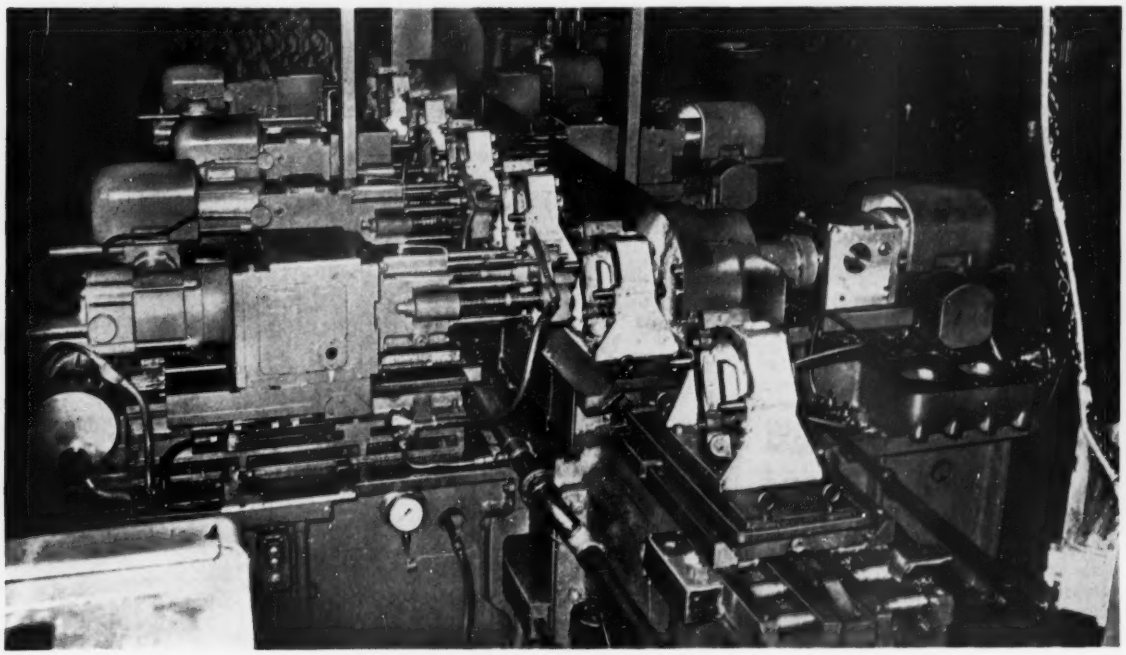

Fig. 31. The transfer machine for the gear carrier

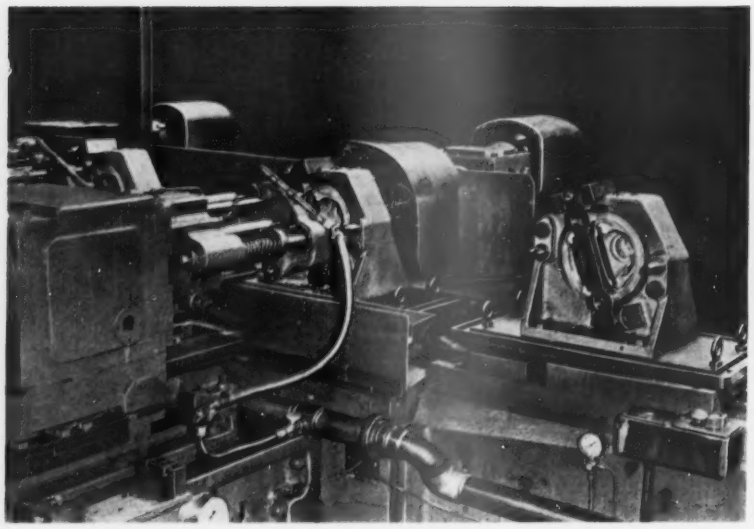

Fig. 32. The loading and first working stations on the machine shown in Fig. 31

Both the eighth and ninth stations have single-spindle heads at both sides of the transfer line. The left-hand head at the eighth station is tooled to ream two holes, while the right-hand head is for drilling a $\frac{5}{8}$ in hole from the rear face through into the forward half of the casting. This is a particularly difficult operation to mount on a transfer machine, since the hole is very long in relation to its diameter and is at an angle to the surface where drilling starts. At the ninth station, the left-hand head carries a tool for boring a diameter to size, while the speedometer hole is tapped from the right-hand head. The tenth and final working station has a multi-spindle head at the left for drilling nine holes in the flange face and one hole in an internal web. A long hole breaking through into the bore is drilled from the right-hand head at this station. This completes the machining on the rear cover, except for drilling one hole on a sensitive drilling machine.

Gear carrier

A particularly interesting machining sequence is employed on the cast aluminium rear axle gear carrier. For the amount of machining that is necessary, the component is handled a remarkably small number of times. The first machining operations are carried out on a Fay automatic lathe with a Skinner three-jaw air chuck arranged for the jaws to grip on chucking pieces at the rear of the flange. A drawbar holds the component back in the chuck.

Four turning tools are mounted on the front tool slide. They rough and finish turn the flange O.D., rough turn a register and rough turn the corners of the bearing cap lugs to a specified diameter. There are five tools, three facing and two chamfering, mounted in the tool box on the rear tool slide. They face the bearing cap joint faces in relation to the flange face, machine the rear face of the flange and form a spigot to size; rough face the front face of the flange; and form chamfers on the front and rear edges of the flange. There is also a facing and forming tool carried on the auxiliary overhead slide to finish the flange face in relation to the bearing cap joint faces and the register.

The casting is then passed to a seven-station Austin in-line automatic transfer machine, see Figs. 31 and 32. Location in the work fixture is taken from the flange diameter and face. A swing bar across the differential bearing lugs is used to clamp the work in the fixture. The swing bar also makes contact in the half bearings to give the correct radial location. In addition, the swing bar has two accurate holes to take guide pillars mounted on the spring-loaded guide bush plates on the machine heads. Additional register between the bush guide plates and the work fixture is afforded by two short pillars on the work fixture, which locate in holes in the bush guide plate.

At the first working station there is a 12-spindle horizontal drill unit at the left. It drills eight holes in the flange and four in the bearing cap joint faces. There is a single spindle boring head at the right for rough boring three diameters. A horizontal multi-drill head at the left of the second working station countersinks the holes drilled at the first station. The right-hand station is vacant.

Two horizontal multi-spindle drilling heads are used at the next station. The head at the left reams the holes in the flange, and that at the right countersinks the other end of the flange holes. At the fourth working station a blow-out and gauging unit at the left clears the tapping holes in the cap joint faces and checks them for depth; the right-hand station is vacant. At the final working station the holes in the cap joint faces are tapped from the left-hand head, while from the right-hand head the end opposite the flange is faced and a radius is formed on it.

At this stage the bearing caps are fitted. After the assembly is completed, the assembled carrier is loaded into a fixture on a Vertimax lathe. Location is taken from the flange diameter and the back face, and clamping is effected by means of manually rotatable clamps pulled back by an air cylinder. Two turning tools, one roughing and one finishing, are mounted on the right-hand head. They rough and finish turn the register diameter on the bearing caps and remove excess material from the corners of the bearing cap lugs. Two chamfer tools in the left-hand head then form chamfers on the top and bottom edges of the register.

From the Vertimax machine, the assembled carrier is transferred to a three-spindle double-end Precimax fine borer. This machine carries a fixture that can be manually indexed through 90 deg. Location in the fixture is taken from the flange diameter and the back face and radially from a reamed hole. The machine has two

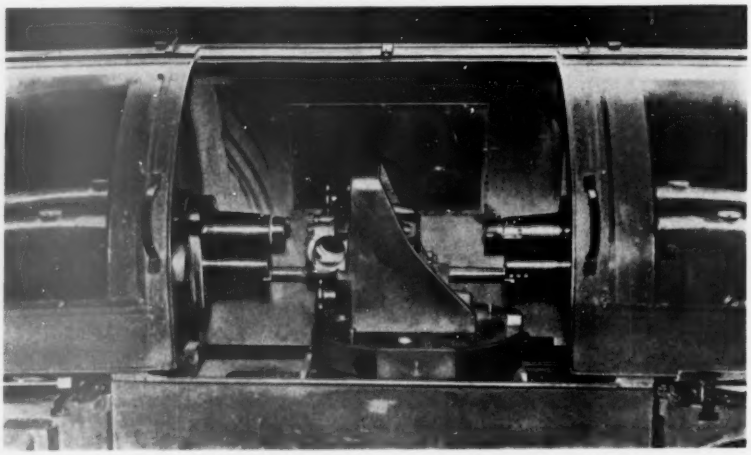

Fig. 33. Precimax four-spindle double-end machine for fine boring the gear carrier

spindles in the left-hand head and one in the right-hand head.

The first machining is carried out from the rear spindle of the left-hand head with the longitudinal axis of the component in line with the axis of the work spindle. Two boring tools and a chamfering tool are mounted in the boring bar. These tools machine the housings for the hypoid pinion shaft bearings. One housing is rough bored and rough faced to depth, another is finished to size and a chamfer is formed on the end of the bore.

When the machining from the first spindle is finished, the work is automatically returned to the central position. It is then indexed through 90 deg to bring the axis through the differential bearing bores into line with the front work spindles. These two spindles are arranged for out-feeding since each of the bearings has a stepped bore. The work is traversed to the left-hand head and feeds over the bar which carries one boring and one boring and facing tool. During the traverse to the left, the smaller diameter is finish bored to size. At the completion of the traverse to the left, the traverse is automatically reversed and continues at feed rate, while the out-feed tool is brought into action to rough bore the second diameter and rough face the bottom of this bore. Traverse to the right continues to carry the work to the right-hand front spindle which carries out the same cycle of operations on the other differential bearing.

Machining on the assembled gear carrier is completed on a four-spindle double-end Precimax fine borer, see Fig. 33. This machine also has a fixture that can be indexed manually through 90 deg, and as on the previous machine, the rear spindles are arranged for machining elements related to the axis of the hypoid pinion shaft bearings, while the front spindles are once again arranged to machine the differential bearings. Location is effected in the same manner as on the three-spindle Precimax machine.

To begin the final machining operations, the work is traversed to the left-hand rear spindle where one bore is finished to size and the bottom of the bore is finish faced in relation to the centre of the differential bearing bores. The work is then traversed to the right-hand rear spindle which carries a boring and facing tool and a boring and chamfering tool. These tools finish two bores to size, machine a face in relation to the centre line of the differential bearings, and form a chamfer at the mouth of the bore.

The fixture is then automatically traversed to the central position and stopped. It is then indexed through 90 deg to bring the axis through the differential bearings into line with the front spindles which are both arranged for out-feeding. The fixture is then traversed to the left and the bearings pass over the boring bar. At the end of the traverse to the left, the out-feed comes into operation and the direction of table traverse is reversed so that the larger of the two bearing diameters is finish bored during the reverse stroke. At a pre-determined point in the return stroke the bar in-feeds to face the bottom of the bore in relation to the axis through the pinion bores. A similar sequence is then carried out from the right-hand front spindle on the other differential bearing. On this component, accuracy not only of individual dimensions but also in regard to relationships between different elements is of fundamental importance. To ensure that the specified close tolerances are maintained, every rear axle gear carrier is checked in a special receiver gauge specially designed to allow all the checking to be carried out quickly and accurately.

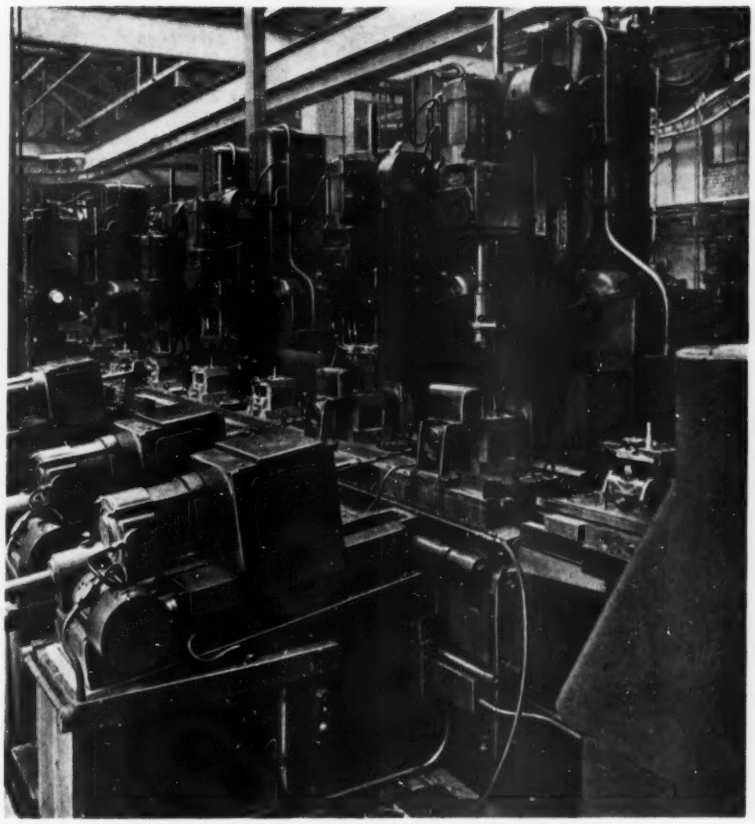

Fig. 34. In-line transfer machine for the swivel axle

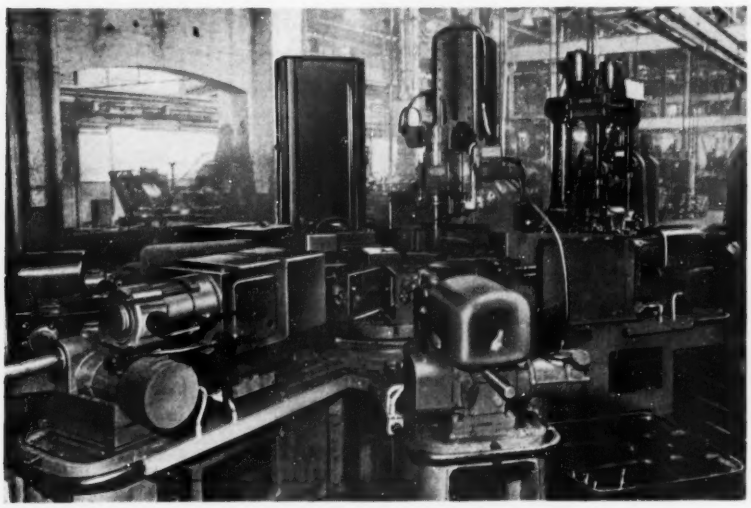

Fig. 35. Rotary transfer machine with Austin unit heads for the swivel axle

Swivel axle machining

The form of the swivel axle is such, and many of the specified tolerances are so close, that a complex machining sequence cannot be avoided. However, although the sequence is complex, the actual number of times the component is handled is relatively small; this has been made possible by the use of three machines, a Ryder Verticalauto, an Austin in-line automatic transfer machine and an Austin multi-station rotary transfer machine.

To begin the machining sequence, the swivel axle is loaded into a centring fixture on a double-end machine. The fixture is arranged to locate and centralize the end of the stem and the swivel pin bosses. On this machine the end of the stem is centred from one end and the lightening hole portion is centred from the other. This gives location points for the next operation sequence, which is carried out on a No. 6 Ryder Verticalauto.

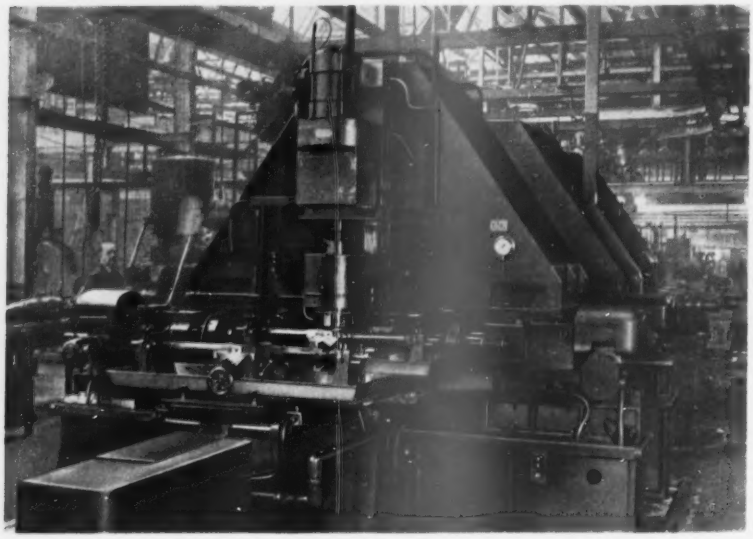

Fig. 36. The loading end of the in-line transfer machine for the rear axle casing

The work-holding fixture has a dead centre that locates in the centred portion of the lightening hole. An air-operated cemented carbide tipped centre locates in the centre in the stem end. This centre is operated independently of the tool slides and is arranged to come down after indexing is completed but before the tools engage the work; it is withdrawn after the tools have been retracted but before indexing starts. To prevent the work from coming out of the jaws during indexing, there are clamps with floating jaws across the ribs.

Loading is effected at the first station on the Ryder machine. Three tools are used at the second station, two turning tools on the right-hand former copying slide and an undercutting tool on the facing slide. The turning tools machine two diameters on the stem and the undercutting tool breaks down the end of the stem for parting off and forms a chamfer.

The turning slide, with two turning tools and a facing tool, and the facing slide, with two facing tools, are used at the third station. Two diameters are rough turned on the stem, a radius is rough formed to blend into a recess in the ribs and the top of the register boss is rough faced from the turning slide. The tools in the facing slide rough machine the flange face, rough form a register and rough machine the front face of a boss.

Only the left-hand former copying slide is used at the fourth station where the taper portion of the stem is turned to size, and a chamfer is formed on the edge of a boss. At the fifth station there are three turning tools on the turning slide and three facing tools on the facing slide. The turning tools finish machine two stem diameters and semi-finish a register, while the tools on the facing slide finish machine three elements. At the final station two turning tools on the left-hand former copying slide finish turn another two stem diameters, while a turning tool on the facing slide finish turns the register to size. The tool on the facing slide is mounted in a caliper-type tool-holder with a ball race that acts as a roller steady.

The stem end is then parted off to length on a facing lathe. Following this, the inner flat faces of the stop leg are milled in relation to the centre swivel pin boss and the face of the bottom swivel pin boss is milled. A side and face milling cutter is used. At the same setting, the lower end of the stop legs and the end face of the top swivel pin are milled with another side and face cutter. The component is then passed to a 16-station in line automatic transfer machine, see Fig. 34.

Because of the component's shape, a fairly elaborate work-holding fixture is necessary. The component is loaded into the fixture with its stem up, with location taken from the diameter of the register and the flange face. Clamping is effected from the underside in such a manner that a vee registers against the bottom swivel pin boss and centralizes this boss with a floating support under the two steering arm lugs. Hand-operated, compensating-type clamps are used.

At the first working station there is a single spindle drill head to the left. It is at an angle of 7½ deg for drilling a hole to a depth of approximately one inch in one of the swivel pin bosses. There are two heads, one horizontal and one vertical at the right of this station. The horizontal head is set at an angle of 7½ deg to the horizontal for drilling a hole to a depth of approximately one inch in the other boss. A Landex die-head is mounted in the vertical head unit for screwing the stem end.

There is a single spindle drill head, inclined at 7½ deg to the horizontal, at the left of the second working station. It drills through the top swivel pin boss at $\frac{1}{4}$ in smaller diameter than the hole drilled in that boss at the previous station. A horizontal single-spindle head inclined at 7½ deg and a four-spindle vertical head are mounted at the right of this station. The horizontal head drills through the bottom swivel pin boss, while the vertical head drills four tapping holes in the flange.

The third is an idle station. At the fourth station there are two heads at the right-hand side, while the left-hand side is vacant. One head, at 7½ deg to the horizontal, counterbores the hole in the bottom swivel pin boss while a four-spindle vertical head is used for counterboring the four tapping holes in the flange. A somewhat unusual tooling is employed on the left-hand head at the fifth station. This head is set at 7½ deg to the horizontal and carries a hollow milling cutter that is used for turning the top swivel pin boss. At the right-hand side of this station there are two heads. One is set at the appropriate angle to the horizontal for rough machining the bottom face of the bottom swivel pin boss. The other, a vertical head, is tooled for reaming the four counterbored holes in the flange. The next is an idle station.

A blow-out and gauging unit for clearing and checking the holes in the flange is mounted at the left of the transfer line at the seventh station, while at the right a head at 7½ deg to the horizontal carries a facing cutter for finish machining the end face of the bottom swivel pin boss. Two heads are mounted at the right-hand side of the eighth station. One is a vertical unit for tapping the holes in the flange; the other is tooled to open out the hole in the bottom swivel pin boss. At the same station, a single spindle head at the left-hand side of the transfer line is tooled for opening out the hole through the top swivel pin boss. The ninth station is idle.

There are three single spindle heads

at the next station. Two heads, one at the left and the other at the right, set at the appropriate angle to the horizontal are used for reaming the swivel pin bosses, while a vertical head drills the lubricator hole in the top swivel pin boss. At the eleventh station, the mouth of each swivel pin boss is chamfered from two heads at 7½ deg to the horizontal, while the greaser boss is pin-faced from a vertical head. The next station is idle. A vertical unit with a milling head at the thirteenth station mills a keyway in the end of the stem. To complete the work on this transfer machine the cotter pin hole is drilled through the stem and the greaser hole is tapped from two heads at the final working station.

A certain amount of machining, chiefly drilling and tapping, must be carried out from the side opposite the stem of the swivel axle. These operations are carried out on an eight-station rotary indexing machine with Austin unit heads. The component is located in the fixture from the flange face and the register diameter, and from one counterbored hole in the flange. It is clamped in position with the stem pointing radially upwards.

Loading is effected at the first station. At the second station, a single-spindle horizontal unit drills a $\frac{7}{16}$ in diameter lightening hole to an approximate depth of 1½ in. This hole is opened out to $\frac{5}{8}$ in diameter for a depth of about ½ in from a single-spindle horizontal head at the third station. The greaser hole is drilled from a horizontal head at the fourth station and is then pin-faced at the fifth station. Two steering arm bosses are also pin-faced at this station. A two-spindle horizontal head at the sixth station carries two counterbore drills for counterboring holes in the steering arm bosses, while a similar head at the seventh station is tooled to ream these holes. Also at the seventh station there is a vertical head arranged for drilling a cotter hole through the stem at right angles to the cotter hole drilled on the in-line transfer machine. To complete the operations on this machine there is a three-spindle horizontal tapper at the eighth station for tapping the greaser holes and two holes in the steering arm boss. This rotary transfer machine is shown in Fig. 35. To complete the machining on the swivel axle, the inside face of each swivel pin boss is counterbored on one machine and two stem diameters are ground on another.

Rear axle casing

As these notes are concerned only with machining sequences, the methods employed for fabricating the rear axle casing will not be discussed in detail. It must suffice to say that the most modern types of welding equipment are used to produce a sound casing with all the various fittings accurately positioned.

When all the welding is completed, the machining sequence proper is started on an Austin unit double-end boring machine. On this machine a cone is formed in the mouth of the axle tube at each end. At the same setting, two diameters are bored in each tube end, one to finished size and one to leave an allowance for subsequent reaming.

From the double-end boring machine, the axle casing is transferred to a dual - drive, double - end Maximatic multi - tool lathe. The component is loaded between centres and gripped by jaws that expand in the bore at each end. This machine carries three tools on each of the front slides for finish turning three diameters on each end. Tools on the rear tool slides finish machine the flange faces to give the correct overall width between the flanges, form a radius in front of the register diameter and face a shoulder. At the same time, a facing and chamfering tool forms a chamfer and faces a shoulder in relation to the flange face, while a chamfering and turning tool chamfers the end of the tube and form turns a diameter for threading. The component is then passed to an eleven-station Austin unit in-line automatic transfer machine.

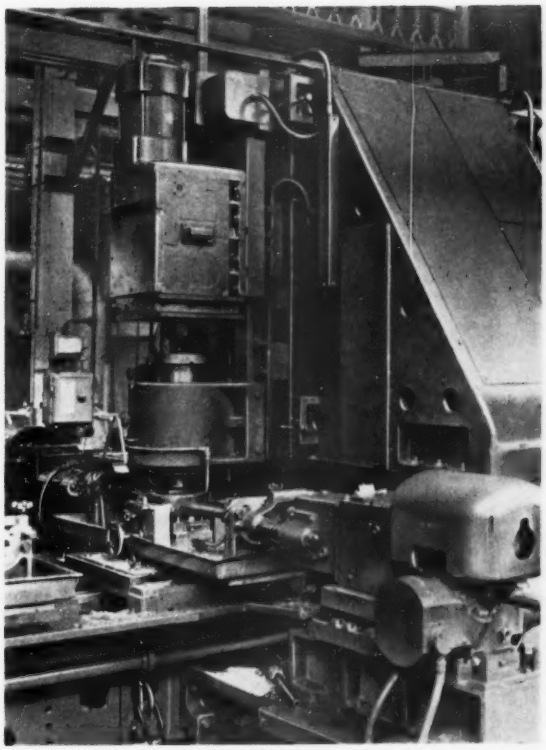

Fig. 37. An intermediate station on the machine shown in Fig. 36

Although in-line transfer machines for rear axle casings have been used for some time in the United States of America, to the best of our knowledge the Austin Motor Co. Ltd. are the first British organization to employ transfer machining on these components. The loading station and the first working station of the machine are illustrated in Fig. 36. The component is located from the flange face

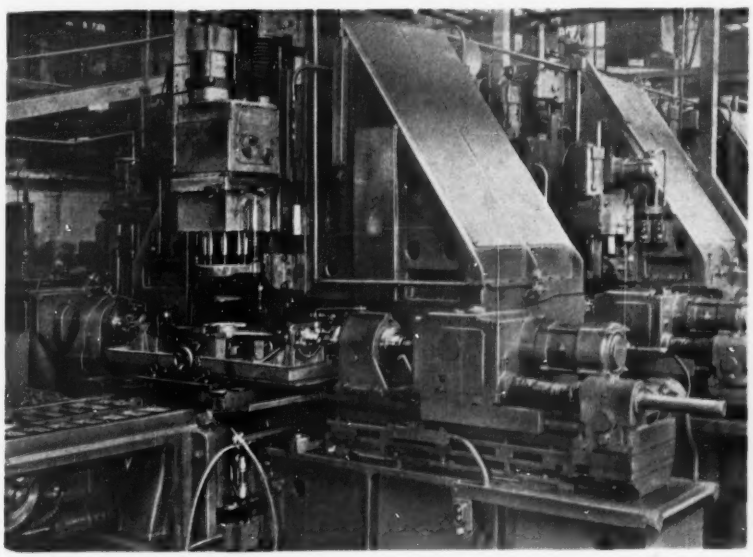

Fig. 38. The final working station on the rear axle casing transfer machine

Fig. 39. One of the transfer machines for gear selector rods

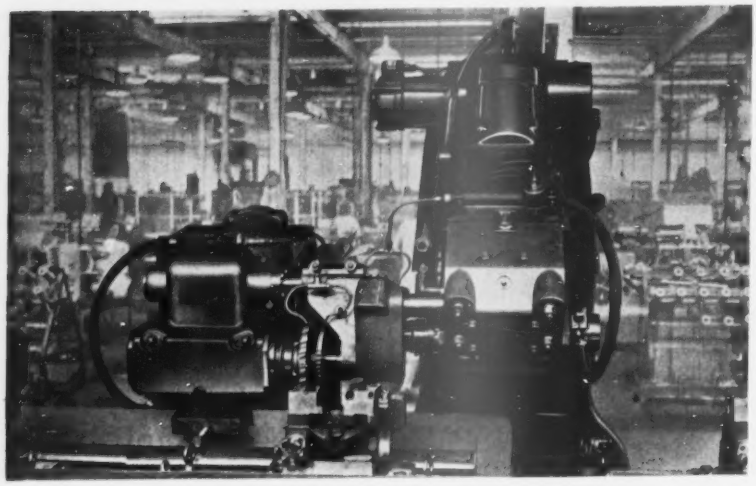

Fig. 40. A milling station on the machine shown in Fig. 39

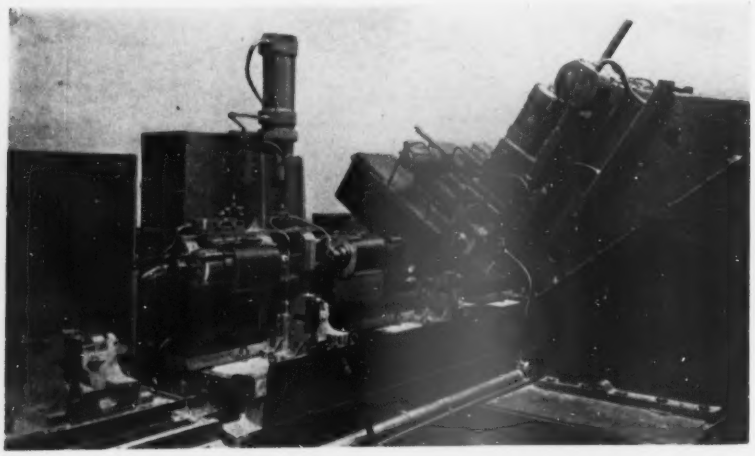

Fig. 41. Centring heads on a selector rod transfer machine

and the register diameter at each end by means of one fixed and one sliding locator. Clamps are fixed on either side of the "bowl," which is supported underneath by adjustable cam-operated jacks.

In certain respects this machine differs from most other transfer machines. Generally, in-line transfer-machines have the machining heads mounted on bases or columns placed at either side of the transfer line, but the considerable overall length of the axle casing makes such an arrangement impracticable for this application. What has been done for this application is to arrange horizontal heads on either side of the transfer line for machining towards the ends of the tubes, while vertical heads for machining the banjo portion are mounted on frames that bridge the transfer line.

At the first working station there is a four-spindle horizontal unit at each side for drilling four holes through the flange. Mounted on the bridge at this station, there is a vertical unit with a special boring head for machining to size two bores in the banjo portion. The next station comes under the bridge frame and is necessarily an idle station. At the next station, horizontal units on either side of the transfer line are used for reaming the flange holes drilled at the first working station. Also at this station there is a vertical head mounted on the bridge frame and tooled for machining the face of the bowl in relation to the centre line of the tubes. This station is shown in Fig. 37.

The next working station has a single-spindle horizontal head at each side of the transfer line. Each head carries a special tap to form the oil return groove thread in one end of the tube. A vertical head mounted on the bridge frame drills eight holes in the joint face. The bores of both axle tube ends are reamed from single-spindle heads at either side of the next station, while the holes in the joint face are countersunk from a multi-spindle vertical head on the bridge frame. At the final working station on this machine, horizontal heads are used to screw each end of the tube and a vertical head is tooled for reaming the holes in the joint face. This station is shown in Fig. 38.

Selector rods

There are three selector rods, and practically the whole of the machining on each rod is carried out on an in-line automatic transfer machine. There is, of course, a transfer machine for each type of rod. One of the machines is illustrated in Figs. 39, 40 and 41. Of these machines, it is sufficient to say that most of the stations are occupied by milling heads for milling the ends of the rod to length, for milling the jaws and for milling slots in the periphery of the rod which, in conjunction with spring-loaded balls housed in the front wall of the gearbox, form the gear locks.

Each of the rod transfer machines has one unusual feature, namely a station at which both ends of the rod are centred simultaneously. One of these stations is shown in Fig. 41. A rise-and-fall double-end unit is mounted on a vertical column at the side of the transfer line. It carries two heads, one with a centre drill and the other with a centre drill and chamfering tool. These heads feed longitudinally towards each other when the whole unit has been traversed down to the correct position.

Rotary transfer machines

Because of their comparative novelty, the automatic in-line transfer machines will probably have greater interest for production engineers than any of the other machining developments for Austin "A30" power units and transmissions. The in-line transfer machine developments should not, however, be allowed to overshadow completely the important part played by specially developed rotary transfer machines, which are widely used in the production of transmission details. In every case the machine has been specifically designed for one component.

For example, there are three such machines for producing speed forks, one for the 1st and 2nd fork, one for the 3rd and 4th fork, and one for the reverse fork. The machine for the 3rd and 4th speed fork is shown in Fig. 42. The table is arranged for double indexing and carries eight fixtures, four that take the component vertically and four taking it horizontally. For the first machining sequence the component is loaded vertically and after one cycle is transferred to a horizontal fixture. All the machining is completed in two cycles. A similar principle is used for the other speed fork machines.

Another interesting rotary transfer machine is used for the gearbox front cover. It is shown in Fig. 43. Before the aluminium die-casting goes to the rotary transfer machine, the joint face is ground on a disc grinder and some milling is carried out. The machine shown in Fig. 43 has a four-station indexing table. There are seven Austin unit heads, but only five are used for any one component. The reason for the additional heads is to allow the machine to be used for either right- or left-hand covers.

The component is loaded into the work fixture and secured by air-operated clamps. At the first station there is a three-spindle horizontal head. One spindle carries a boring bar with two boring tools and a chamfering tool; each of the other spindles carries a flat-bottomed drill. From this head the main bore and the ball race recess are rough machined, and a chamfer is formed on the mouth of the bore by the tools in the boring bar, while the flat-bottomed drills form shallow recesses. Also arranged to operate at the second station are two angular single-spindle heads; only one head is used, depending upon whether the cover is right- or left-hand. This bores a tapping hole through the lugs.

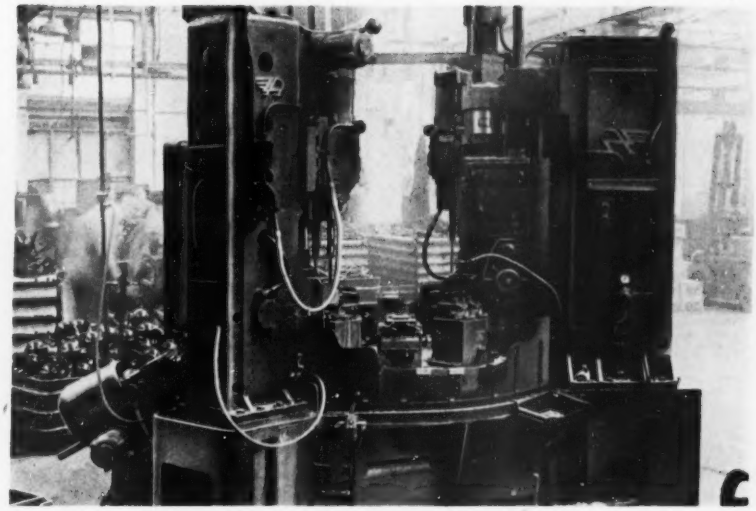

Fig. 42. Austin rotary transfer machine for gear forks

At the third station there is a horizontal unit with a speed reducing head that carries a special tap for cutting the oil return groove. The arrangement of the heads for the fourth station is similar to that at the second station, except that there is a two-spindle in place of the three-spindle horizontal head. At this station the elements rough machined at the second station are finish machined from the horizontal head, and the angular hole drilled at the second station is tapped from one of the vertical angular heads.

There are some components for which rotary transfer machines with the table arranged to index round its vertical axis are not altogether satisfactory. For some such components, Austin unit machines are used in conjunction with a trunnion-mounted fixture arranged to index about its horizontal axis. A machine of this type is shown in Fig. 44. It has an eight station trunnion, two horizontal three-spindle units and two vertical two-spindle units for machining the layshaft.

Loading is effected at the first station, and the second is an idle station. At the third station each end of the shaft is centred from a spindle in one of the horizontal heads. The second spindle in each of these heads comes into operation at the fourth station where a $\frac{5}{8}$ in hole is drilled to a depth of $\frac{7}{8}$ in in each end of the shaft. At the fifth station, the third spindle in each horizontal head continues the drilling at each end at $\frac{3}{4}$ in diameter

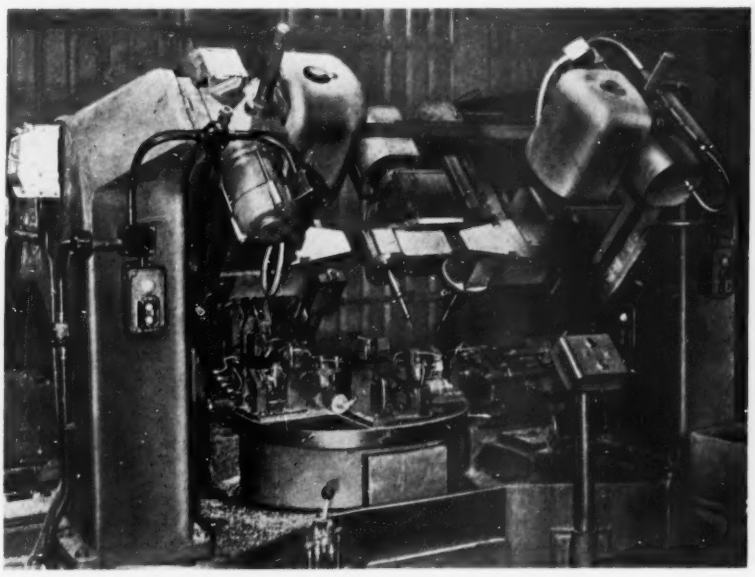

Fig. 43. Austin unit-head rotary transfer machine for gearbox front cover

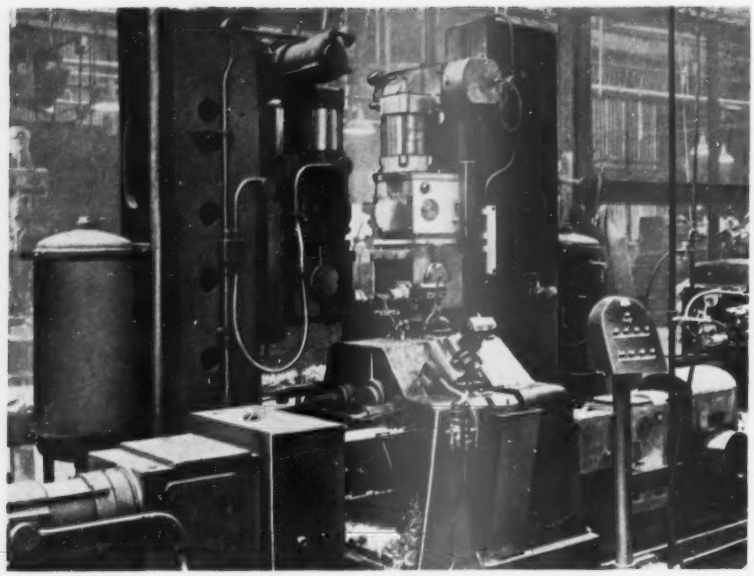

Fig. 44. Austin unit-head machine with eight-station trunnion-mounted turret

to a depth of $1\frac{11}{32}$ in. The vertical units come into action at the sixth station to complete the machining by milling the end step down to the centre line and by drilling the cross holes.

Generally, the speeds and feeds employed are reasonably high, but not in any way remarkable. This is not surprising, since the Austin Motor Co. Ltd. do not make the rate of metal removal the sole criterion for determining machining conditions. On the contrary, the general practice is to work at something appreciably lower than the maximum possible rate. In other words, the machining conditions are selected to give the best possible combination of output and tool life with a balanced production. Although this may cause the floor-to-floor times for certain operations to be significantly longer than the shortest possible, the unit cost may be appreciably lower than would be achieved at higher rates of metal removal. In other words, the production processes for the "A30" power unit and transmission have been scientifically planned to combine efficiency with economy.

There will be general agreement that the developments described in these notes are clear evidence that The Austin Motor Co. Ltd. are prepared to meet the very keen competitive conditions that already exist, and which become even keener, in both the home and export markets. The decision to design and manufacture unit machine heads for the Company's own special purposes was not one to be lightly taken, but the results already achieved have proved that it was wise.

SCANDINAVIAN SERVICE

Conditions under which Road Vehicles Operate

MANUFACTURERS concerned with the Scandinavian markets for road vehicles, and more particularly for commercial vehicles, must give thought to the very severe operating conditions. During the summer, conditions are not significantly different from those encountered in this country, but in winter, and particularly during the thaw, road conditions may become very difficult. In addition, it may be necessary to operate over almost deserted roads, where help, if needed, would scarcely ever be forthcoming.

Winter temperatures are, of course, very much lower than those experienced here, and there is a wide variation over the whole peninsula. For example, in 1951 the lowest temperatures recorded in various parts of Sweden were 19·5 deg F in the extreme south, 8·5 deg F at Gothenburg, −15 deg F at Stockholm lattitude and −43 deg F in the extreme north. Even lower temperatures are experienced in Finnish Lapland, where −40 deg F, combined with strong winds, is a common occurrence. These temperatures make great demands not only on the engines and vehicles themselves, but also on fuels, lubricants and batteries.

Many main roads in Norway and Sweden are metalled, but most of the other roads are surfaced in gravel and sand, and few roads in Finland are metalled at all. Dust is a problem on the gravel roads. To minimize the dust nuisance, the principal gravel roads are sprayed with calcium chloride or sulphite lye during the summer. These binding media are sometimes mixed with clay spread out on the road and moistened. The result is acceptable in dry weather, in rain one or two inches of salty mud may be formed.

In dry weather, the dust puts great demands on engine air cleaners. The gravel, and even the concrete and asphalt roads can cause much trouble, especially in the spring, owing to the effects of thawing. On slippery winter roads, the large American cars with over-powered engines find considerable difficulty, since neither accelerator nor brakes can be used and chains are of little help. The oil bath type of air-intake filter is the only one recommended. Double filters should always be used for diesel engine fuel injection pumps.

The mixture of gravel, dust, water, clay and calcium chloride or sulphite lye can easily damage chassis bearings and moving parts. Brake drums must be effectively shielded, as must fuel injection pumps and electrical equipment for underfloor engines. The chassis should have anti-corrosion treatment, and the frame should be sturdy yet flexible, particularly for bus or lorry chassis. Low centre of gravity is essential.

To reduce fuel viscosity in winter, it is not unusual to mix the oil with from 20 to 30 per cent paraffin. In the colder areas the fuel tank is often warmed by exhaust gases, by means of a copper coil. Condensation of water in the fuel tank is a frequent source of trouble since it freezes and blocks the piping. For diesel engines a pint of de-natured alcohol is an effective preventive against this.

Particularly with direct injection engines, the engine temperature cannot be controlled by thermostat when the air temperature falls from 15 to 5 deg F, but good results have been obtained through the use of radiator shutters. Starting difficulties are common and separate engine heaters are sometimes used. These reduce the viscosity of the lubricating oil. The heating of buses is also a problem. Heat from the cooling system was generally sufficient with pre-combustion chamber engines but with direct injection the amount of heat available is insufficient. Separate heating devices are therefore necessary.

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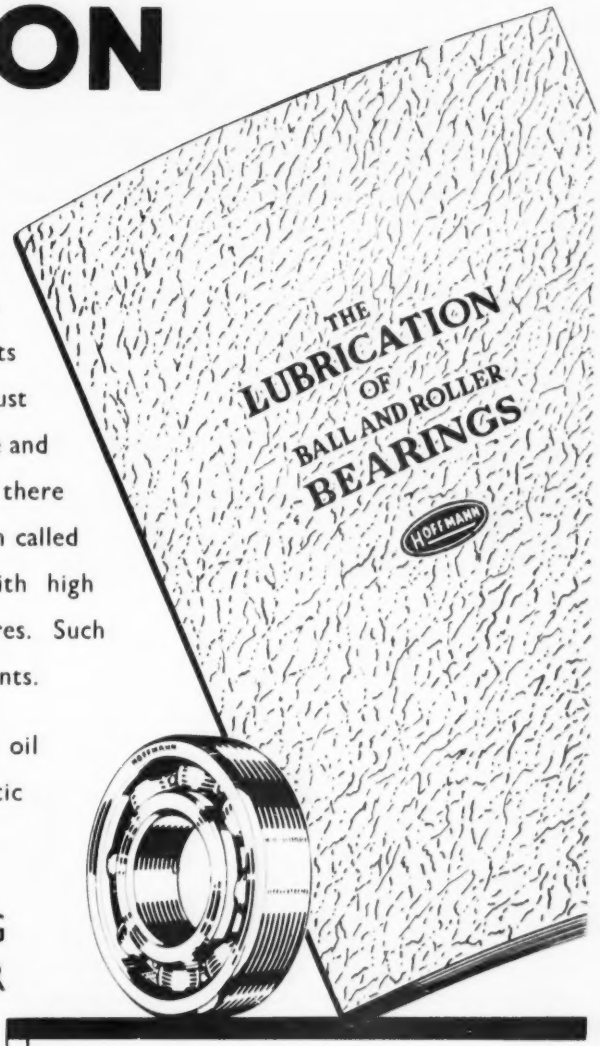
This booklet, a technical paper, deals with both oil and grease lubrication and provides characteristic bearing mountings with lubrication systems.



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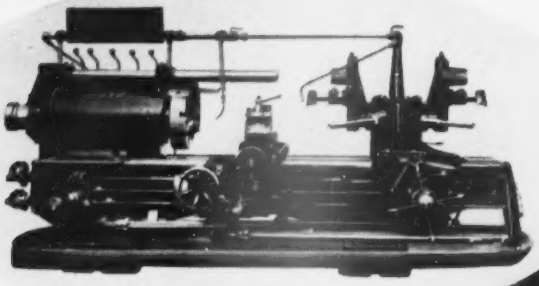

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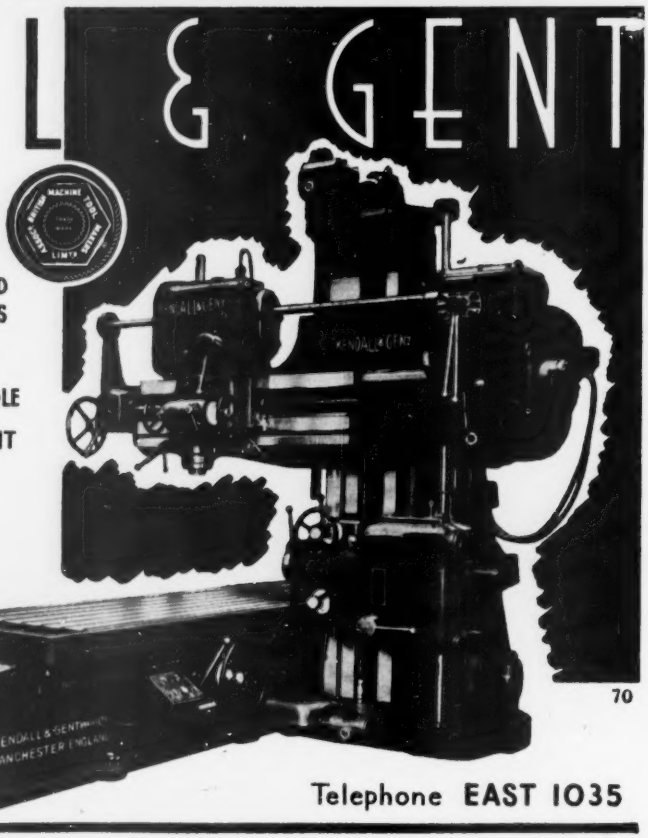

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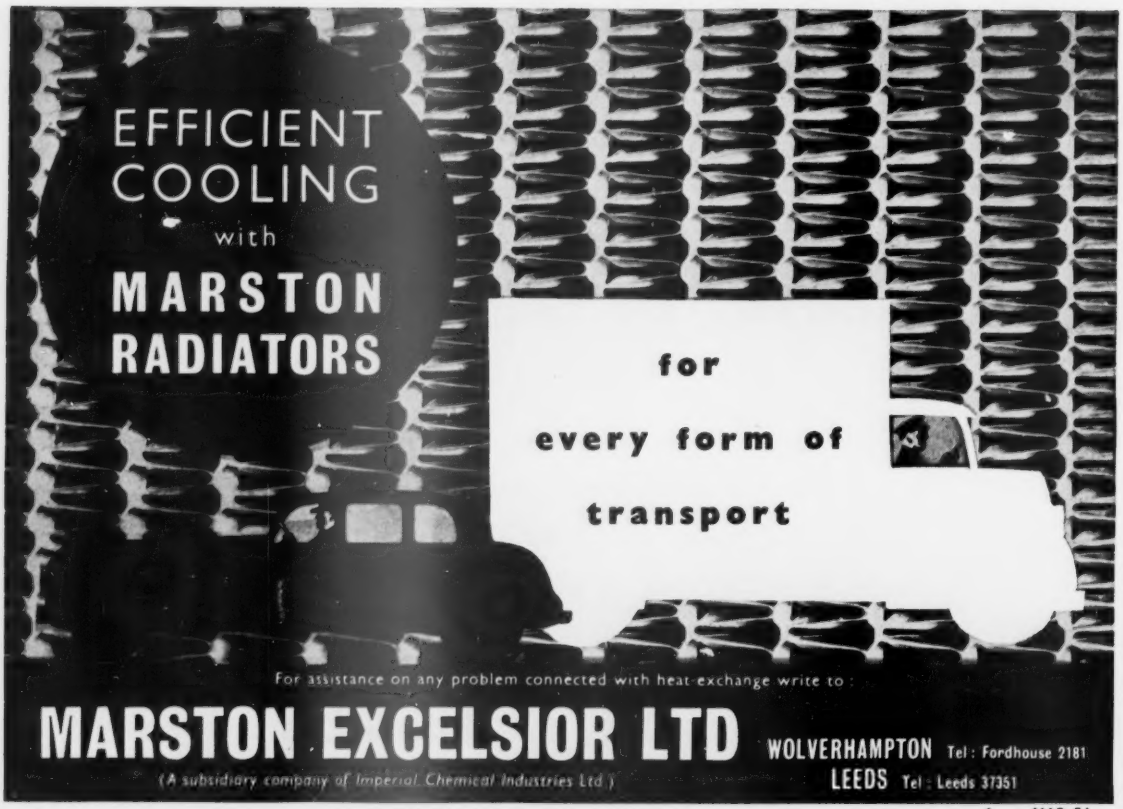

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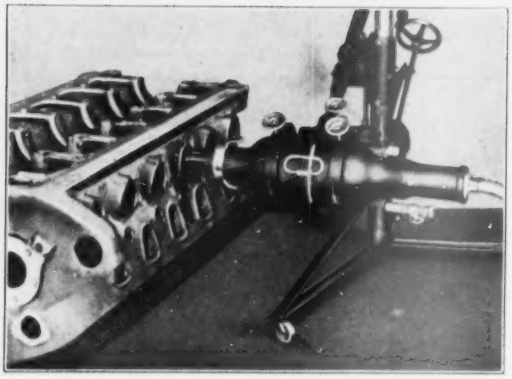

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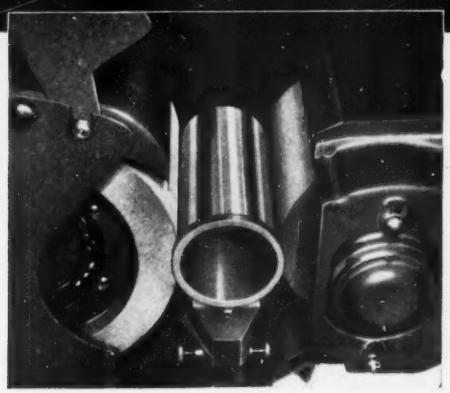

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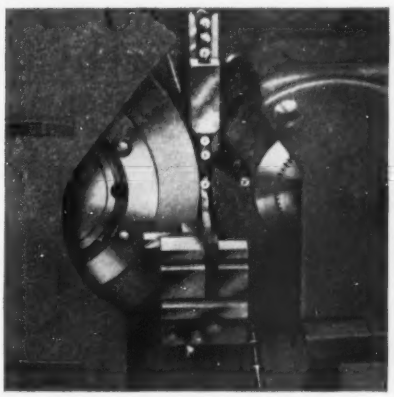

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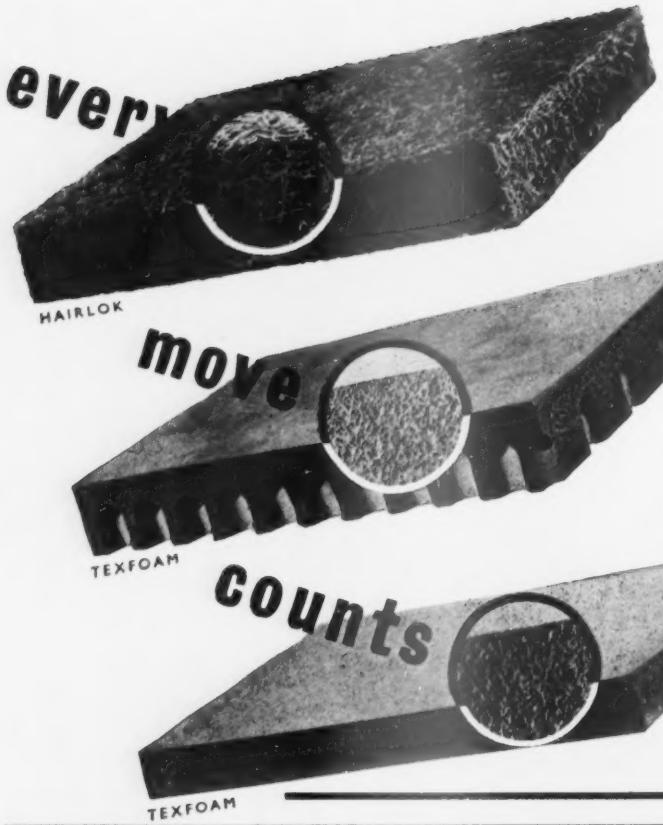

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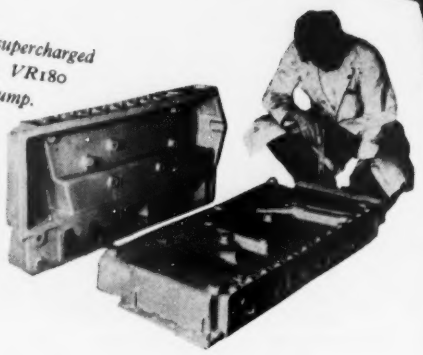

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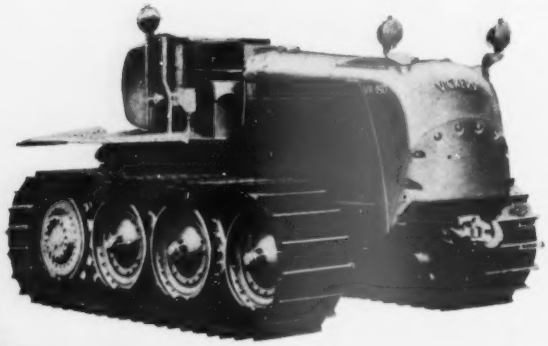

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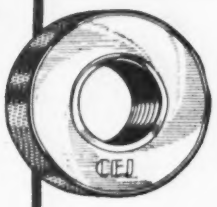

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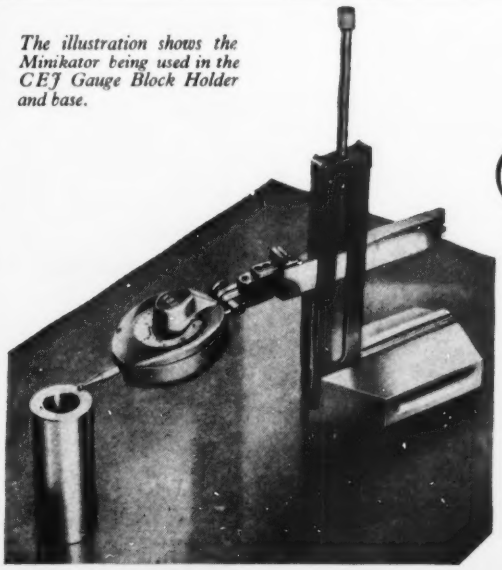

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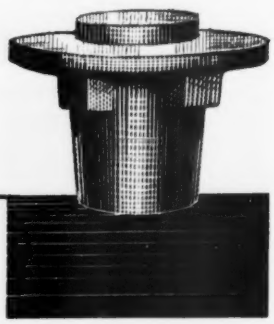

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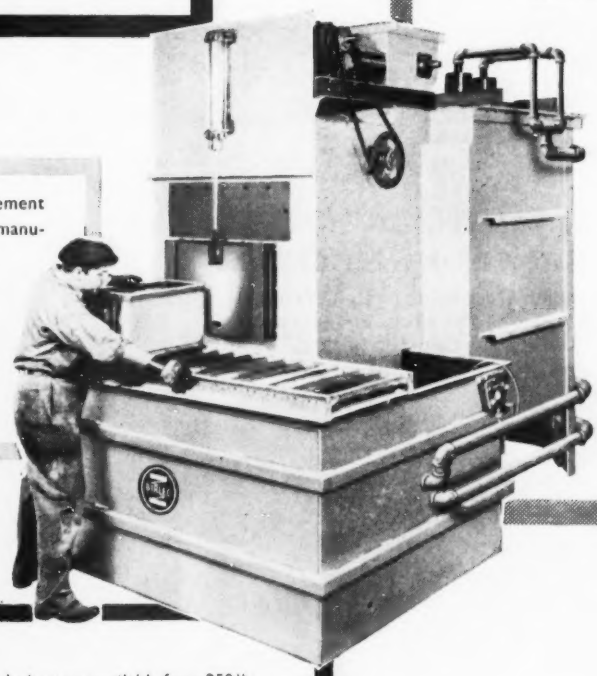

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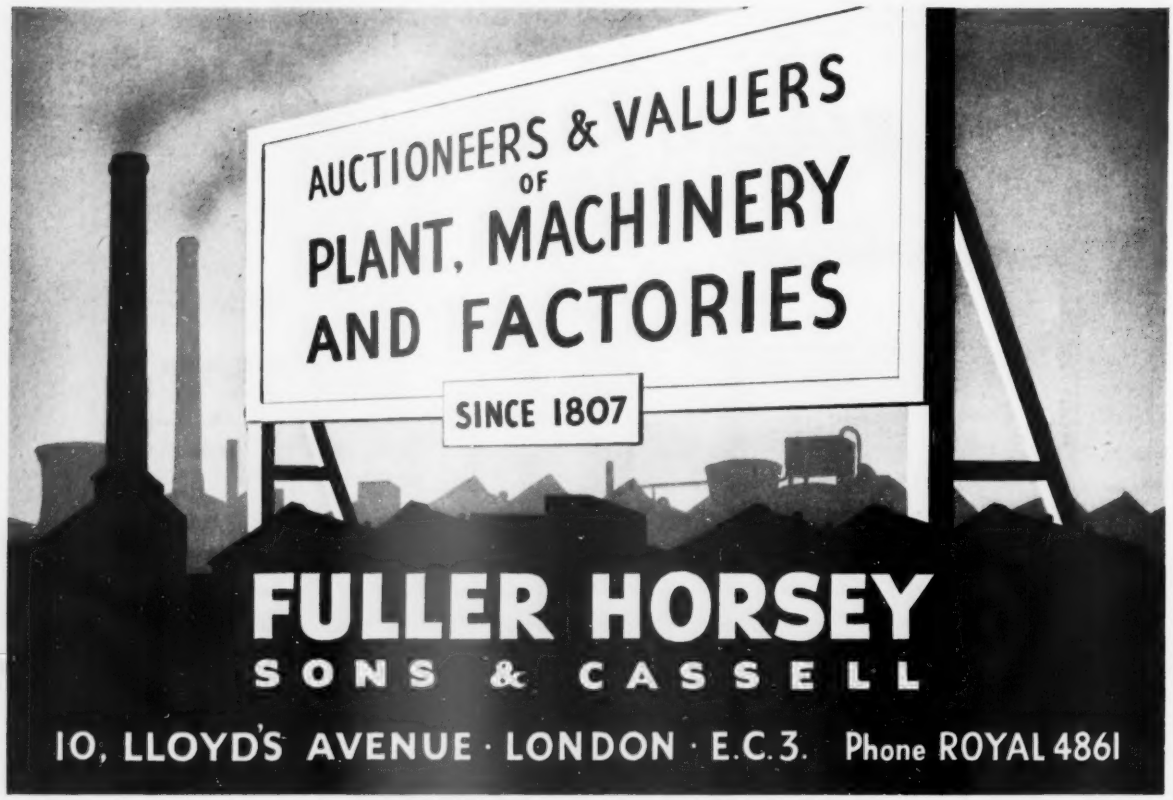
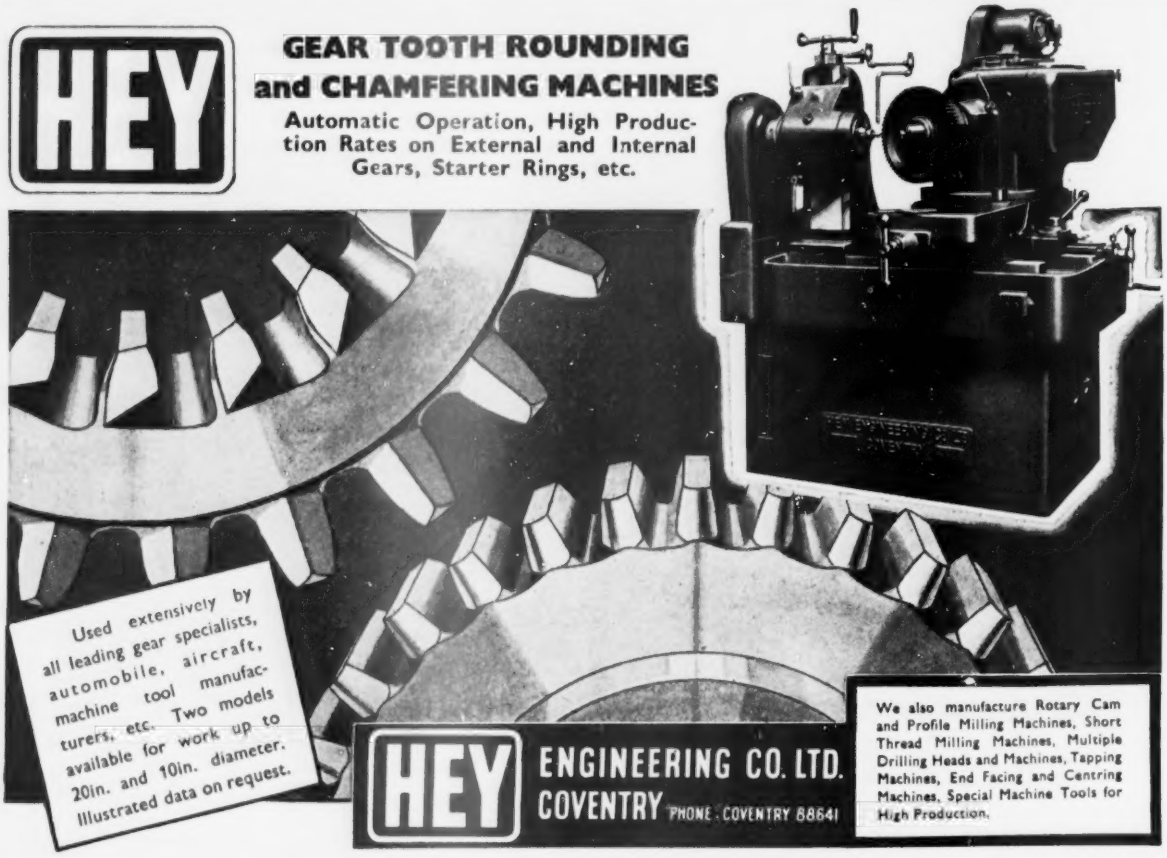


Rathbone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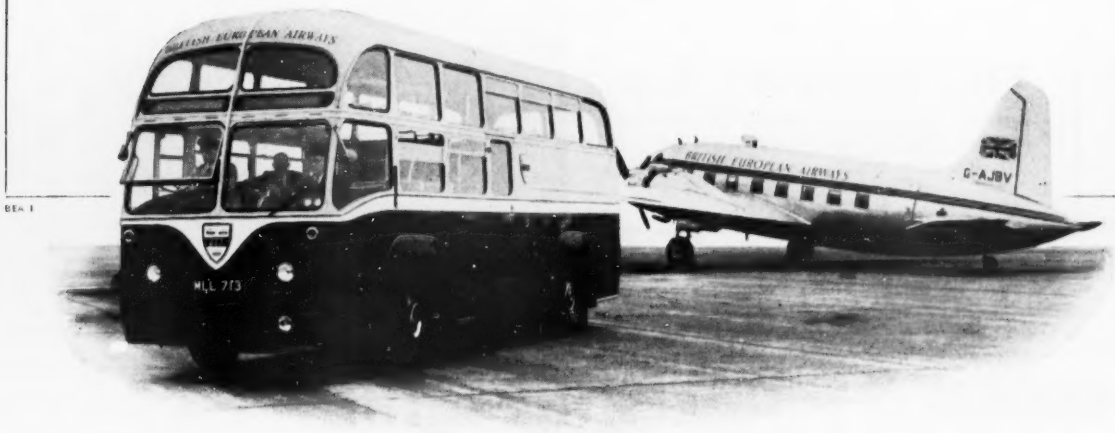

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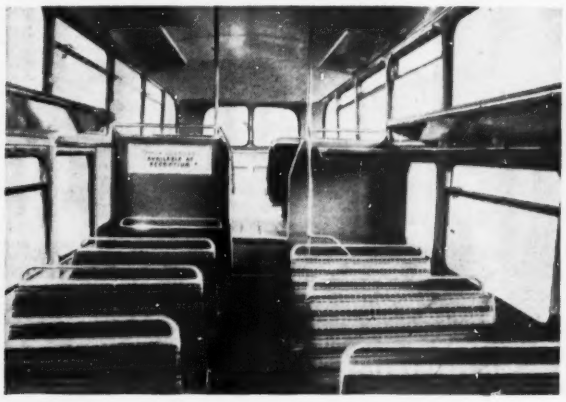

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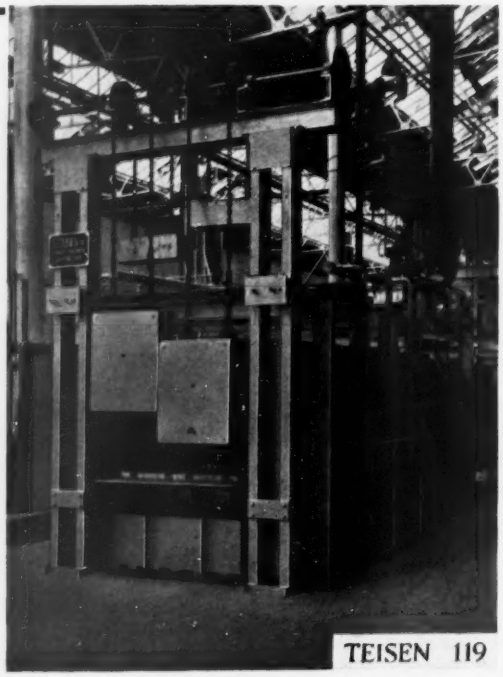

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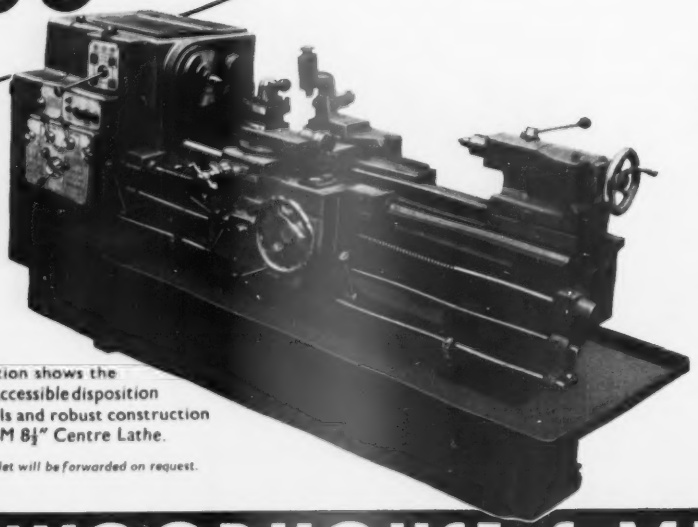

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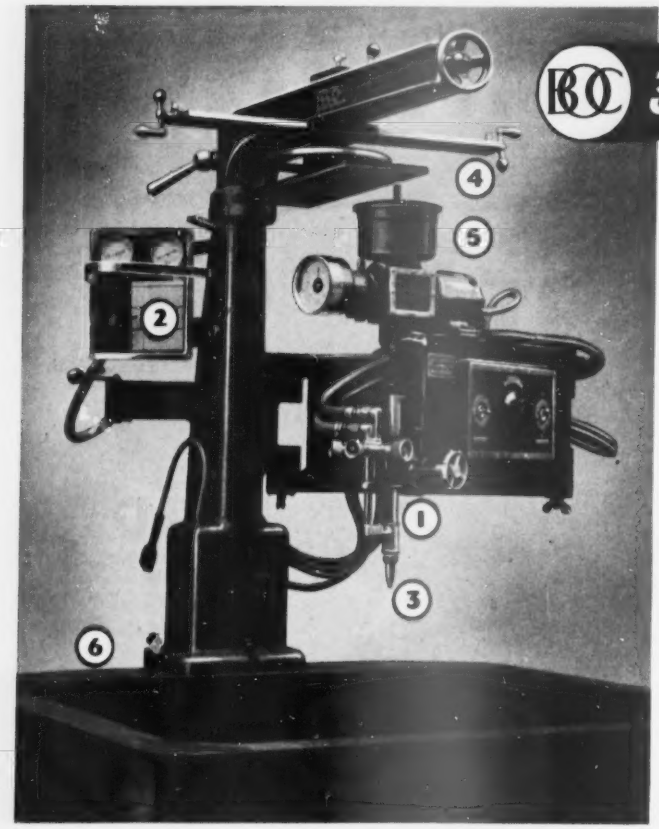

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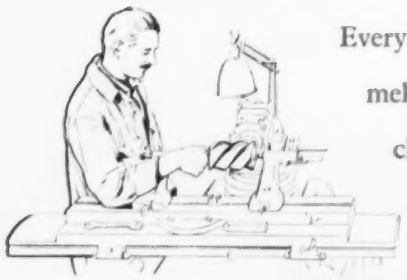

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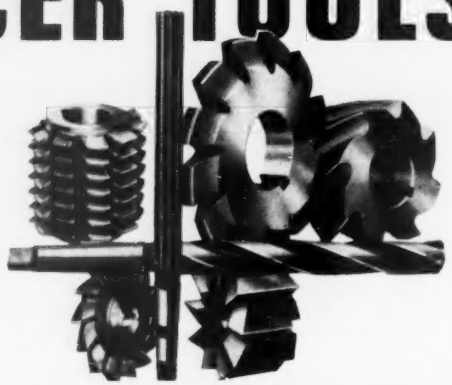

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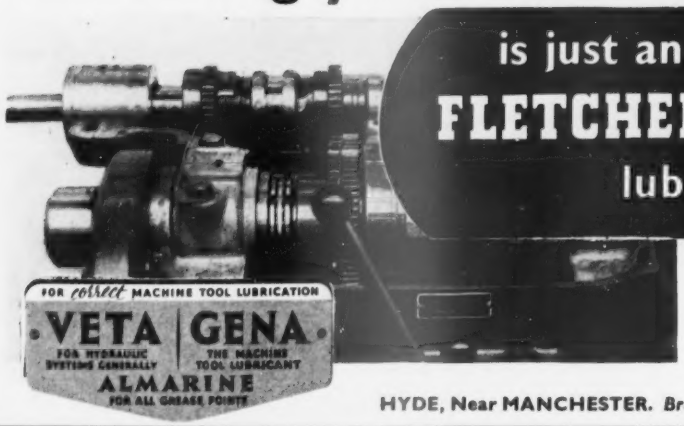

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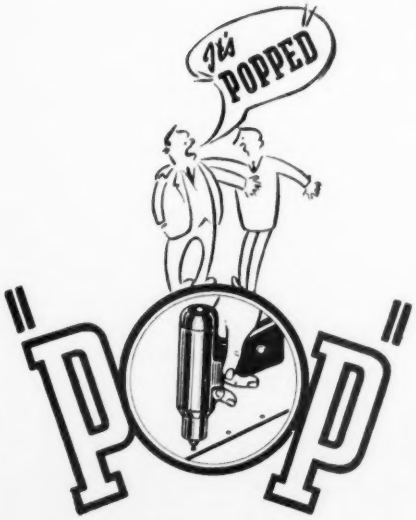

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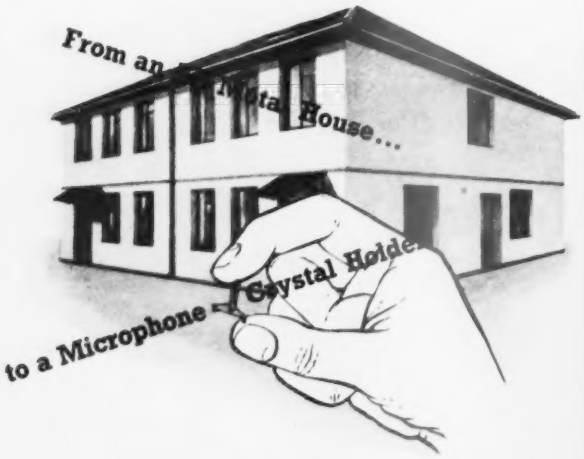

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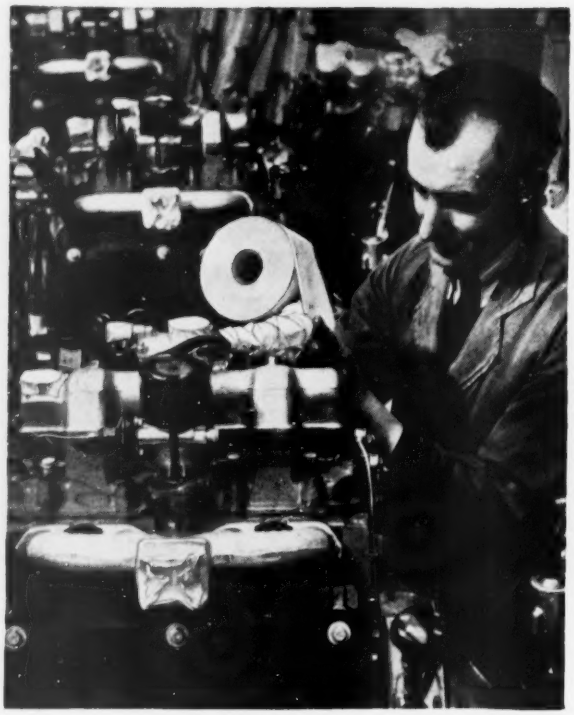

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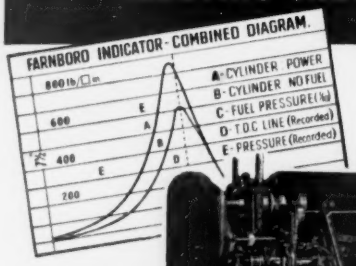

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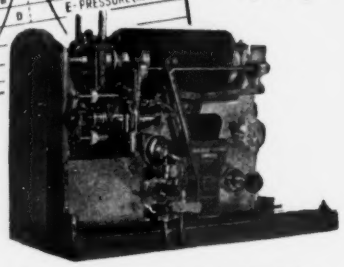

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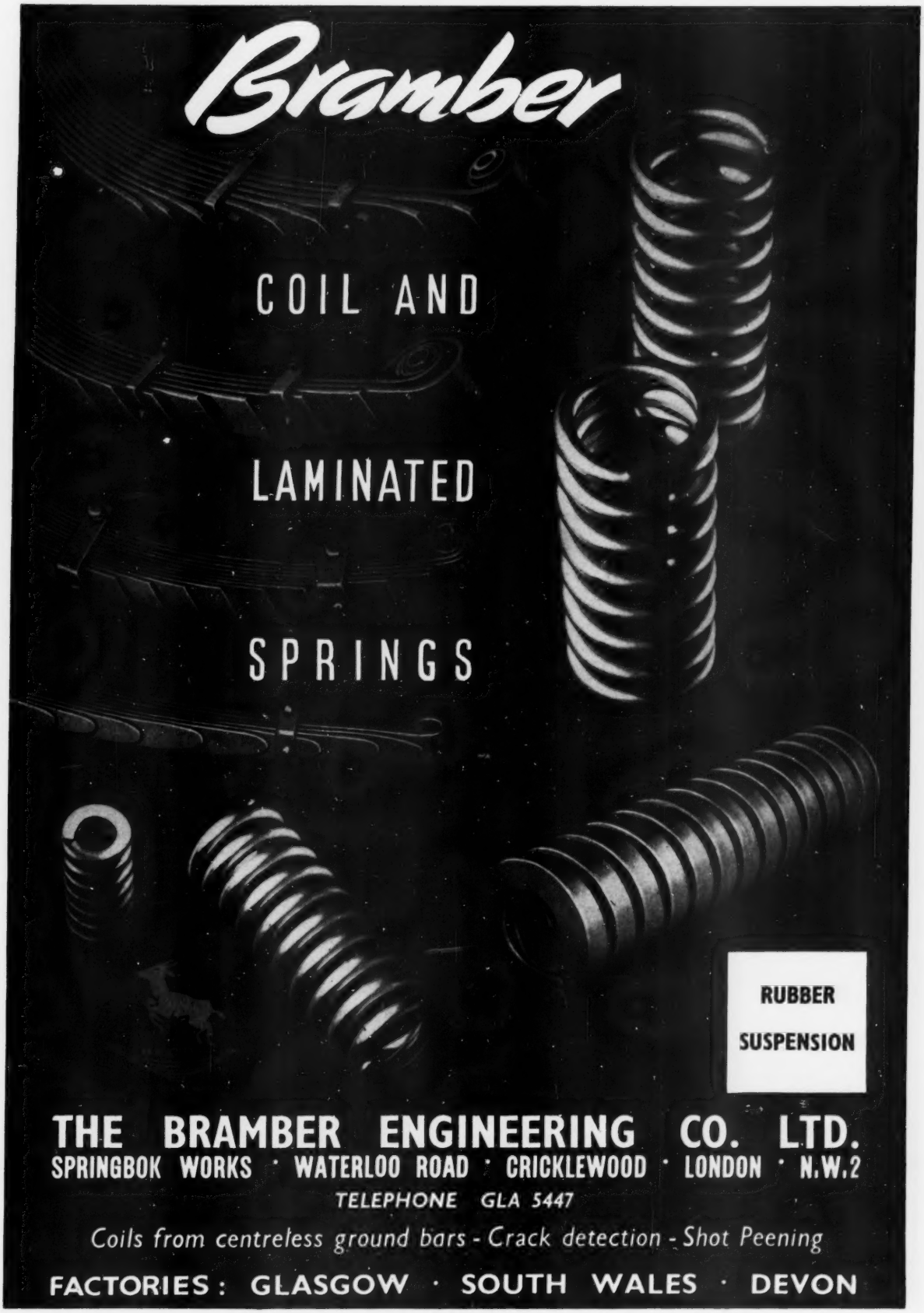
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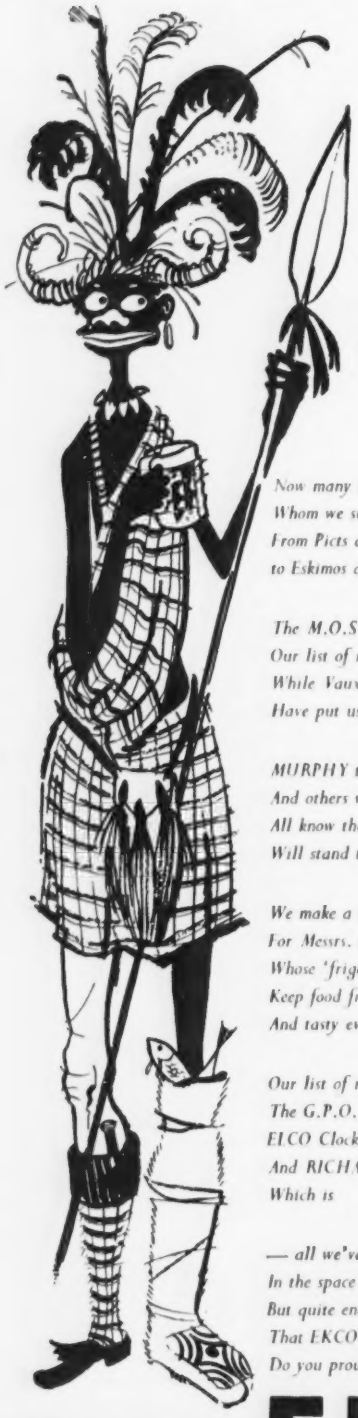

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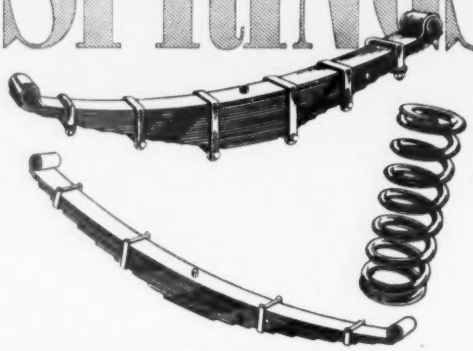

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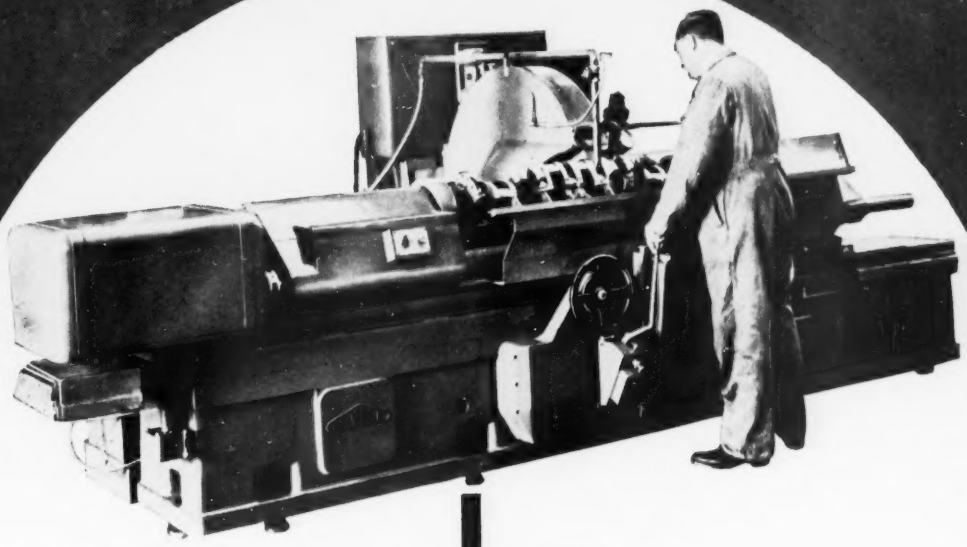

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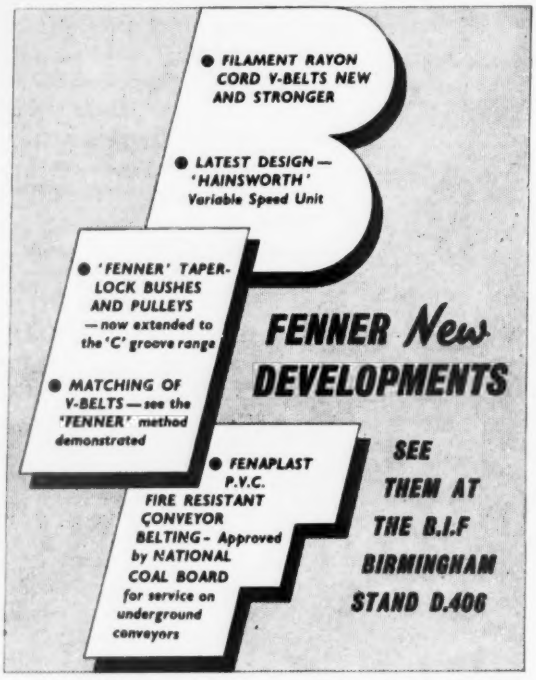
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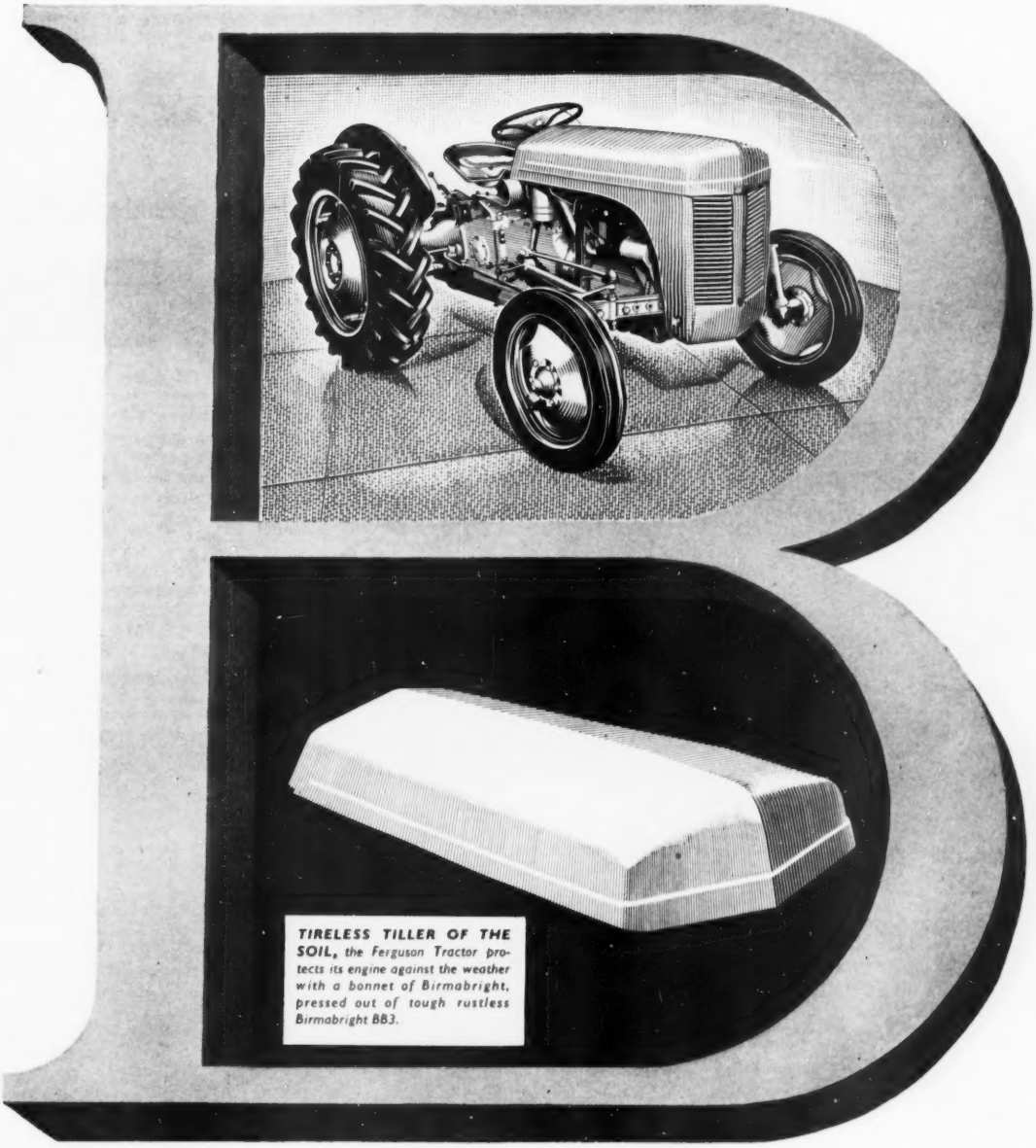

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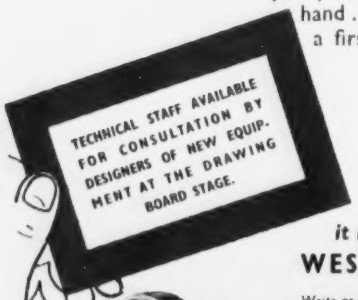

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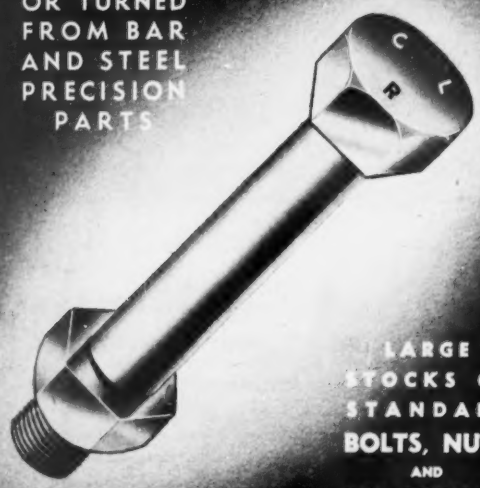

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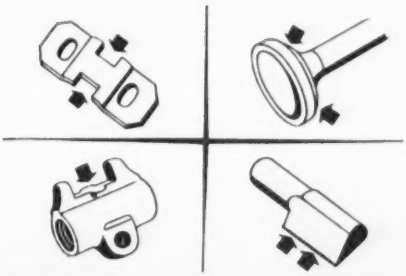

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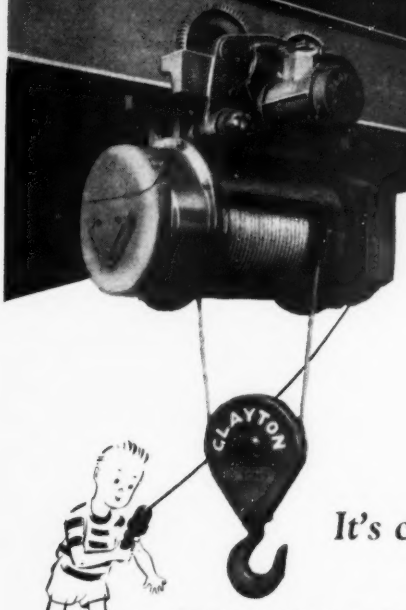

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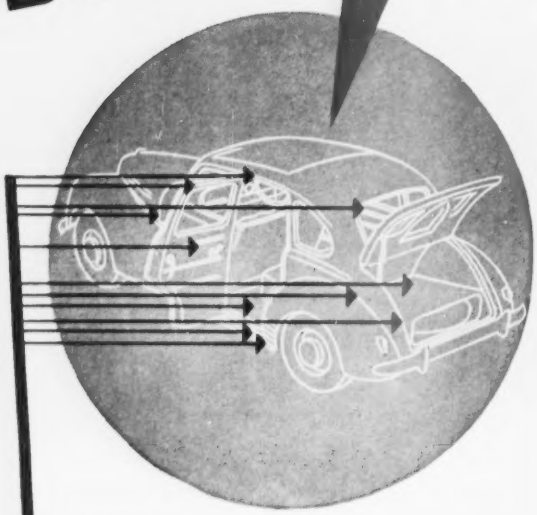

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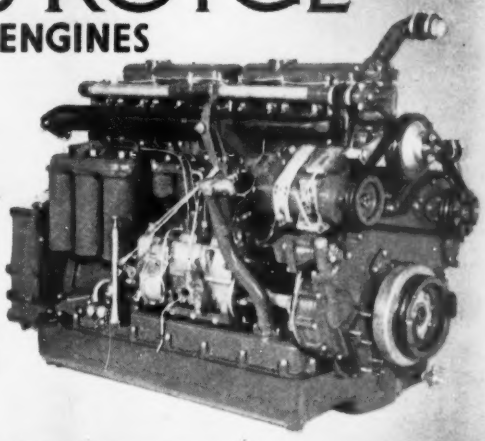

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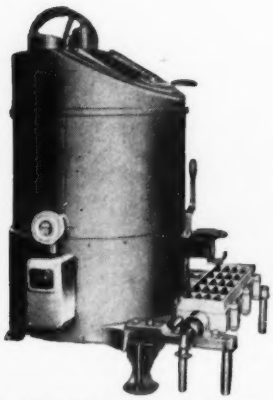

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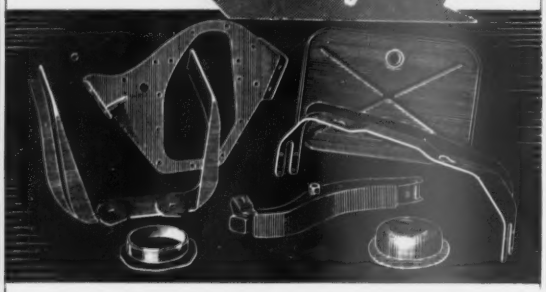

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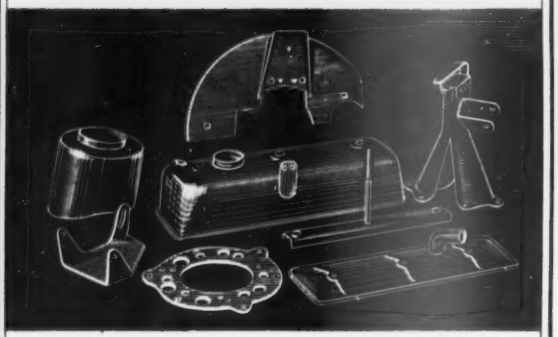

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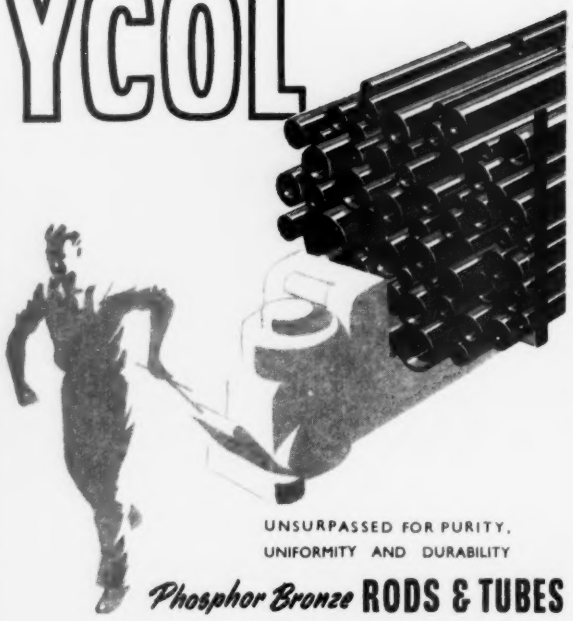

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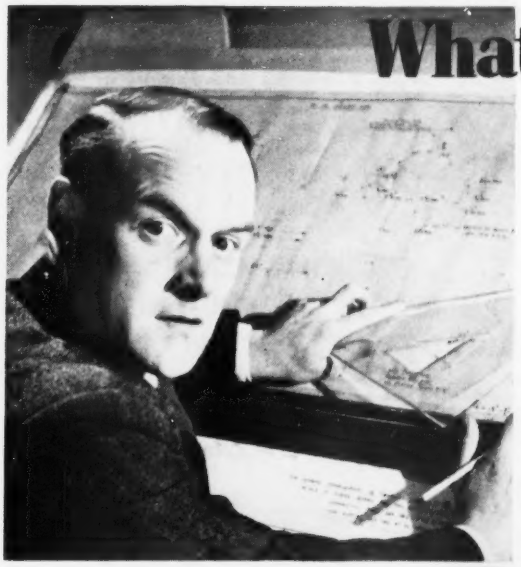

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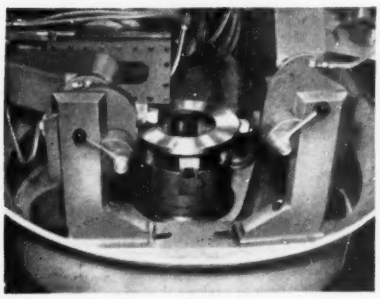

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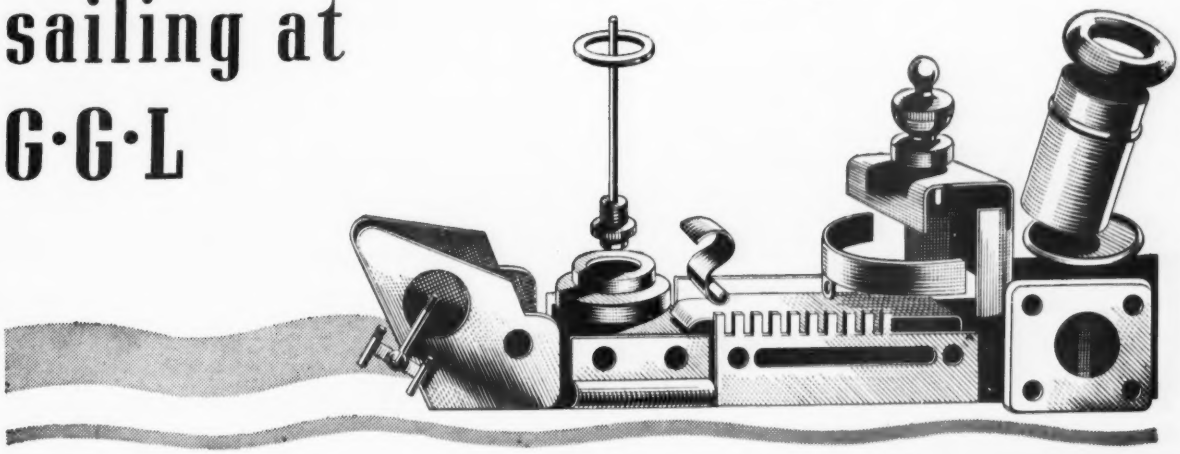

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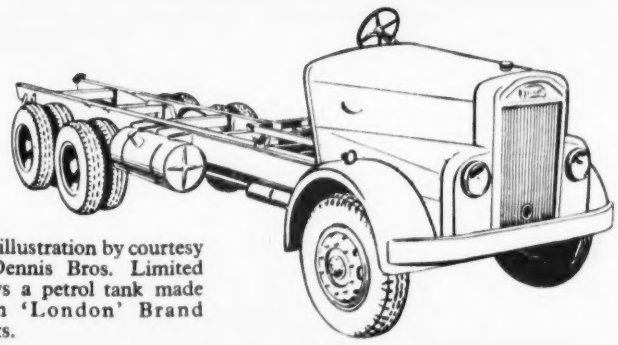

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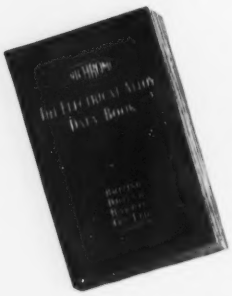

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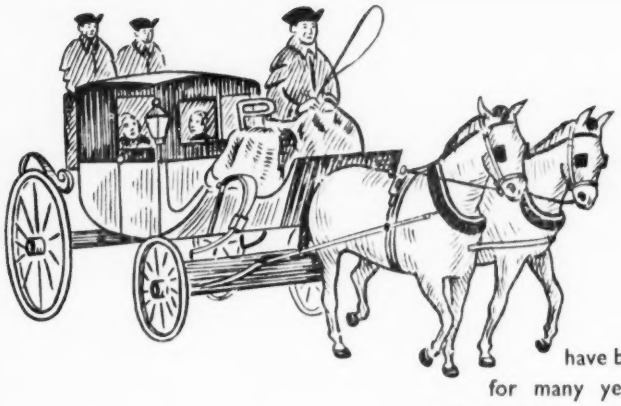

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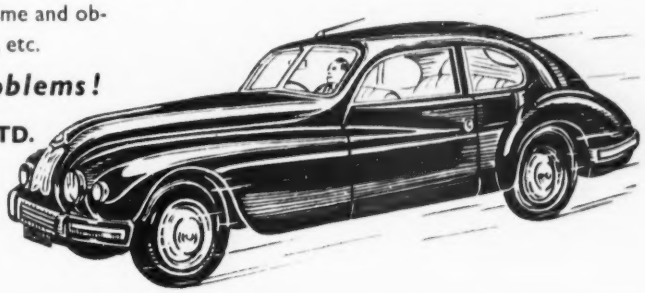

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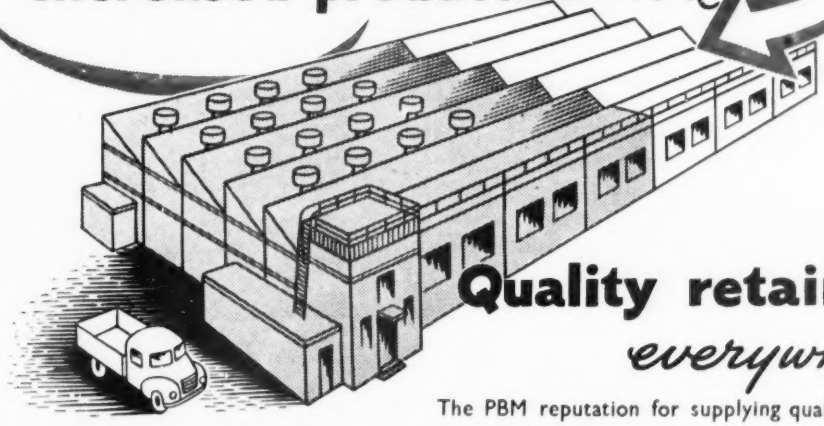

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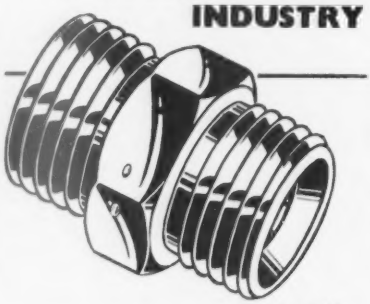

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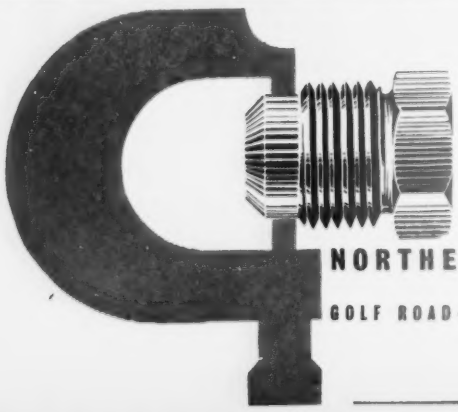

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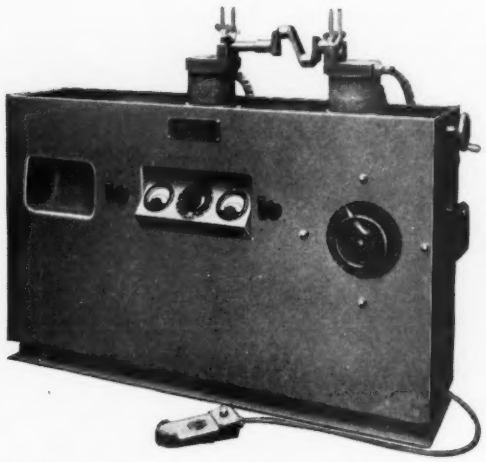

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